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Stress Analysis of Closure Bolts for Shipping Casks

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ABSTRACT

This report specifies the requirements and criteria for stress analysis of closure bolts for shipping casks containing nuclear spent fuels or high level radioactive materials. The specification is based on existing information concerning the structural behavior, analysis, and design of bolted joints. The approach taken was to extend the ASME Boiler and Pressure Vessel Code requirements and criteria for bolting analysis of nuclear piping and pressure vessels to include the appropriate design and load characteristics of the shipping cask. The characteristics considered are large, flat, closure lids with metal-to-metal contact within the bolted joint; significant temperature and impact loads; and possible prying and bending effects. Specific formulas and procedures developed apply to the bolt stress analysis of a circular, flat, bolted closure. The report also includes critical load cases and desirable design practices for the bolted closure, an in-depth review of the structural behavior of bolted joints, and a comprehensive bibliography of current information on bolted joints.

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PREFACE AND ACKNOWLEDGEMENTS

This report contains recommended procedures, criteria, and formulas for the stress analysis of closure bolts for shipping casks used for transporting radioactive materials. The work, funded by Transportation Certification Branch, within the Office of Nuclear Material Safety and Safeguards of the U.S. Nuclear Regulatory Commission (NRC), took place at Lawrence Livermore National Laboratory (LLNL). D. Tiktinsky was the NRC Project Manager and H. W. Lee was the NRC Technical Monitor for this project. Recommendations set forth are the results of applying existing knowledge and ASME Code to the special design conditions of shipping casks.

The authors had discussions with a number of experts in bolted joints and wish to thank H. W. Lee of the NRC for the overall technical guidance; T. Lo, and M. W. Schwartz (retired) of LLNL for conducting the initial investigations; A. Blake (retired) of LLNL, J. H. Bickford of the PVRC (Pressure Vessel Research Council) Subcommittee on Bolted Flanged Connections, and T. Sawa of Yamanashi University, Japan for stimulating discussions on the subjects of bolt preload, bolt prying, and bolt forces. The authors also wish to thank the following LLNL staff: M. Sands and B. Smith for editing, M. Carter, D. Halaxa, and S. Murray for word processing.

EXECUTIVE SUMMARY

The procedures, criteria, and formulas developed in this study are recommended for the structural analysis of closure bolts for shipping casks used for transporting radioactive materials. The recommendations result from applying existing knowledge and industrial codes for bolted joints to the special design conditions of shipping casks. The special conditions include the consideration of large, flat closure lids with metal-to-metal contact within the bolted joint, high fire temperatures, severe impact loads, and strict leakproof qualities. To deal with these special conditions, the study explored the bolt prying action, the interaction of bolt preload and applied loads, the limit on bolt deformation, and fracture toughness.

The study concluded that the fracture toughness of bolt materials should meet the ASME Boiler and Pressure Vessel Code (Section III) requirements for bolting materials of Class 1 nuclear power plant components. The bolt deformation should be elastic and the bolt stresses should not exceed the material yield condition. Interaction of bolt forces should include all bolt forces and moments and should be properly combined.

In the study, approximate formulas were specified or derived for calculating bolt forces generated by all regulatory (normal and hypothetical accident) transportation loadings. Results of additional studies conducted for assessing possible prying and bending effects on closure bolts include the development and verification of simplified models and formulas for calculating the maximum prying bolt force and the maximum bolt bending moment in a bolted closure with a flat circular lid. Verification used both experimental and analytical results. Experimental results came from the literature on bolted joints, and analytical results were obtained using sophisticated finite element computer programs and models. The formulas for calculating bolt forces generated by various transportation loads appear in ten tables. (See table of contents for their page numbers.) The derivation of some of these formulas and the background information on the structural behavior of bolted joints are in the appendices.

The presented information shows that for shipping casks, the tensile axial force and the transverse shear force are the primary bolt forces which have the potential to cause catastrophic bolt failures by a single application of the forces. The bending moment plays a secondary role and can produce catastrophic bolt failures only after repeated applications of the moment. The torsional moment is significant only if a torque wrench is used for preloading the bolts. In addition to the existing preload, thermal expansion and prying can generate significant tensile axial bolt forces. Impact and thermal expansion can produce significant shear bolt forces.

Three stress analyses and their requirements and criteria are specified along with methods to facilitate them. These analyses are, namely, the maximum stress analysis of normal transport conditions, the fatigue stress analysis of normal conditions, and the maximum stress analyses of accident conditions.

Suggested ways to minimize bolt forces and bolt failures for shipping casks are an important part of this study. The following are some examples:

- Protect the closure lid from direct impact to minimize bolt forces generated by free drops.

- Use materials with similar thermal properties for the closure bolts, the lid, and the cask wall to minimize the bolt forces generated by fire accident.

- Apply a sufficiently large bolt preload to minimize fatigue and loosening of the bolts by vibrations.

- Lubricate bolt threads to reduce the required preload torque and to increase the predictability of the achieved preload.

- Use a closure lid design which minimizes the prying actions of applied loads.

- When choosing a bolt preload, pay special attention to the interactions between the preload and the thermal load and between the preload and the prying action.

The present studies have demonstrated the following useful information for accomplishing the last of the preceding suggested actions: A flat closure lid of one uniform thickness can produce a greater prying action than a lid with two different thicknesses; and a preload can enhance the prying action of an applied load.

1.0 INTRODUCTION

1.1 Background

A bolted closure can be a weak link in the containment system of a shipping cask for spent fuels and high level radioactive materials. The structural integrity and leakproof qualities of the bolted closure depend on the number, strength, and tightness of the closure bolts. For the safe performance of shipping casks, appropriate methods and criteria were developed for the design and analysis of bolted closure joints. Unfortunately, this effort was hampered by the complex structural behavior of the bolted joint. Appendix I describes some of the complex interactions found among the different components of the bolted joint. The behavior of bolted joints varies significantly with the design and application of the joints. For this reason, the data found in the literature on bolted joints can often appear confusing or even conflicting, and they should not be applied indiscriminately to the evaluation of the bolted closure joints without proper consideration of the differences in design and application. Existing studies and industrial codes (Refs. 1–3) focus on bolted structural joints, pipe joints, and pressure vessel joints which have quite different designs and loadings from shipping cask bolted closure joints. A shipping cask must be designed for significant fire and impact loads, and a large, flat, closure lid. Prior to this study no established standards existed for the design and analysis of bolted closures for shipping casks.

1.2 Scope and Objective

In view of the needs described above, analytical methods and criteria are developed here for the evaluation of shipping cask closure bolts. The methods and criteria pertain only to the closure bolt, not the bolted joint. Although the closure bolt dominates the behavior of the bolted joint, the structural integrity and leakproof quantities of the bolted joint depend on other joint components (i.e., the closure lid, the cask wall, and the gasket (or seal)). One must analyze these components to confirm the structural integrity and leakproof qualities of the bolted closure joint. Further guarantee against leaks may require a combined program of analysis, testing, and maintenance.

Specifically, this report deals with the bolt stress analysis of a circular, cylindrical, cask with a flat, circular, closure lid (as depicted in Fig. 1.1) and describes an acceptable method and criteria for this analysis. As far as possible, closed-form, approximate formulas are developed and presented to facilitate the analysis.

1.3 Approach

The present analysis method and criteria required a review of existing literature and engineering practices or codes regarding bolted joints to identify the significant structural behaviors that are consistent with shipping-cask-closure designs and loadings. Appendix I shows the results of this study. Based on this information, simplified analysis models have been developed to describe these behaviors. In turn, these models have been used to derive approximate closed-form formulas for the quantitative prediction of the resulting bolt forces/stresses from the joint and load parameters. The verified adequacy of the formulas is based on test data and/or finite element analysis models which are more sophisticated and realistic than the simplified models. As shown in Appendix II, the stress analysis requirements and criteria established here are similar (but not identical) to those of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (B&PVC), Section III, Subsection NB for bolted joints of Class 1 nuclear power plant components (Ref. 3). The stress limits have been set on the basis that the bolt material is ductile and the overall bolt deformation remains elastic under normal operation loads.

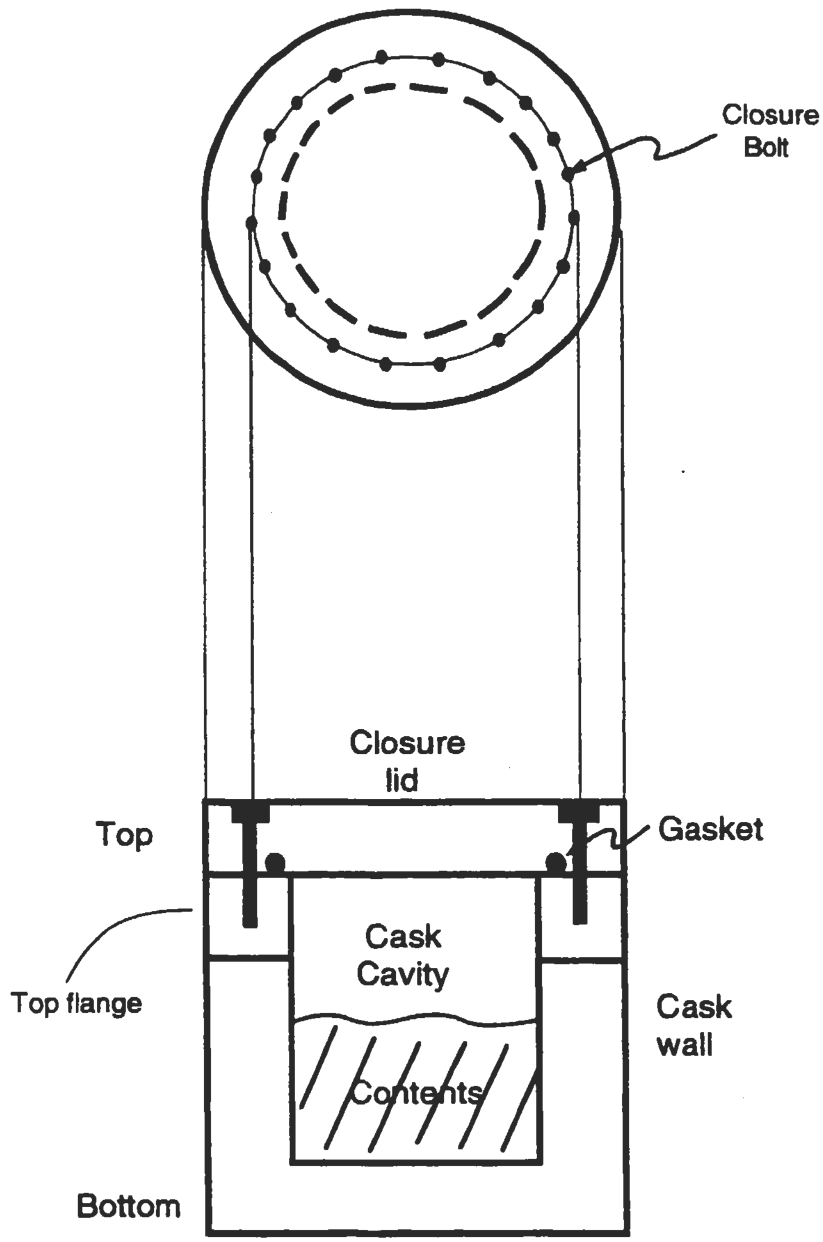


Figure 1.1 Shipping cask showing closure bolt positions.

2.0 BOLTED SHIPPING CASK CLOSURE DESIGNS AND RELATED EFFECTS

2.1 General Geometry

The methods described in this report have been developed specifically for the bolted closure design shown in Fig. 1.1. The flat, circular lid of the closure is bolted to the cask wall using only one row of identical tap bolts which are uniformly distributed along a circle near the lid edge. The bolt circle and the lid edge form concentric circles. Figure 2.1 shows the closure design details considered here. As pointed out in Appendix I, closure design details can significantly affect the forces and moments in the closure bolt. Discussion of these details and their possible effects on the bolt forces/moments appears in following subsections.

2.2 Bolted-joint Design and the Effects of Bolt Bending and Prying

All of the detailed bolted-joint designs shown in Fig. 2.1 have direct, metal-to-metal contact in the joint area between the closure lid and the cask wall. As discussed in Appendix I, when the closure lid is bent under load, a relative rotation may appear between the closure lid and the cask wall. This rotation, in turn, may generate in the closure bolt a bending moment and a prying force. It should be pointed out that the prying force and bending moment are in addition to the bolt forces and moments which the applied loads on the closure lid generate directly or which support the applied loads. (See Section 4 for a discussion of these directly-generated bolt forces for all applied load conditions and the formulas for their evaluation.)

The combined effect of bending and prying is not simple to analyze. However, as Appendices III and IV show, the finite element analysis study reveals that the interaction between the prying and bending actions is weak and an adequate estimate of the bending and prying effects on the bolt can be made by considering the effects separately.

The studies in Appendices III and IV also show that the bending effect is insignificant compared to the possible prying effect. In the sample closure designs analyzed in these appendices, the maximum bending stress never exceeds 20% of the total average tensile stress, whereas the tensile stress attributed to the prying action can be greater than 60% of the total average tensile stress in the closure bolt. This result suggests that the bending stress is not likely to cause large global plastic deformation over the entire cross-section of the closure bolt, but it can still cause local plastic deformation leading to the failure of the bolt by incremental plastic deformation and fatigue. The prying stress remains a potential cause for all possible failure modes of the closure bolt. Comparing the two stresses, the prying stress has the characteristics of a primary stress which is defined in Section III of the ASME B&PVC (Ref. 3) as a stress that can cause a catastrophic structural failure by a single application of the stress, whereas the bending stress is closer to a secondary stress which can cause a catastrophic failure only after repeated applications of the stress.

The studies in Appendices III and IV bring forth two other facts concerning prying and bending effects which have a significant implication in closure bolt design and analysis:

- Both the prying and bending effects can be greatly reduced by the stiffening or thickening of the closure lid. A closure lid thickness which is adequate for supporting the applied load may not be sufficient to avoid a significant prying effect. Accordingly, assessing the possible prying effect is an essential step in closure bolt analysis.

- The maximum prying force usually occurs when the applied load is equal to the preload. Therefore, the bolt preload must be set apart from a critical applied load to minimize the prying effect of the critical load.

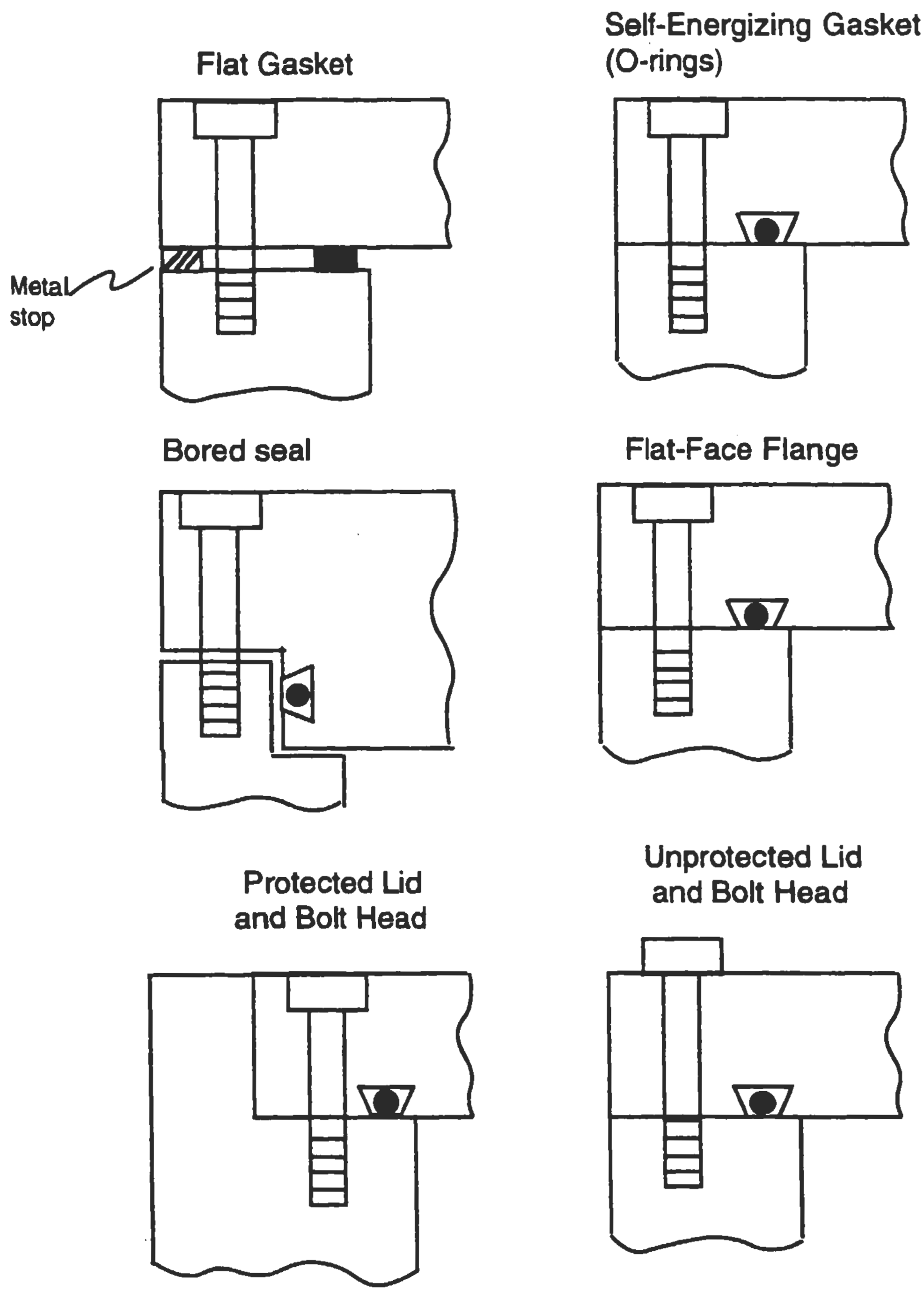


Figure 2.1 Closure designs considered in this report.

In view of the potential importance of prying and bending effects, simplified models and formulas are developed here for the analysis. Appendix III describes the development and verification of two simplified analysis models and formulas, namely the plate-ring and the plate-plate models for the determination of the prying bolt force. Appendix IV presents similar information for the maximum bending bolt moment. Several finite element analyses with various degrees of realism are used to verify these simplified models and formulas.

The simpler of the two formulas for the calculation of the prying bolt force (i.e., the plate-ring model) appears in Table 2.1. Table 2.2 presents the formula for the bending bolt moment. In these formulas, the applied load is generically expressed in terms of the fixed-edge force Ff and the fixed-edge moment Mf which the applied load generates in the closure lid at the bolt circle (assuming that the lid is totally fixed at the bolt circle). The formulas for Mf and Ff are given in Tables 4.1 through 4.8 for all the cask loads that may have appreciable bending and prying actions.

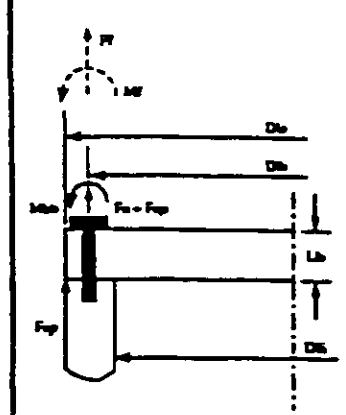
Appendices III and IV show that the formulas listed in Tables 2.1 and 2.2 tend to overpredict the results by a considerable margin because they have ignored the cask wall flexibility and other effects. The main advantage of the formulas is their simplicity; the results can be quickly obtained by hand calculation using these formulas. More precise results can always be obtained by modifying the simplified formulas to include the omitted effects or by using a detailed finite element analysis. However, the decision regarding a finite element analysis should be made with the full awareness that the analysis of the bolted joint is a highly nonlinear problem whose accurate solution can only be obtained by an experienced user with an adequate model and a proven computer program for this type of analysis. The nonlinear finite element analysis results reported in Appendices III and IV were obtained only after a long series of sensitivity studies to determine the proper value to use for the load step and convergence limit.

2.3 Gaskets and Gasket Loads

ASME B&PVC (Ref. 3), Section III, Appendix E, divides gaskets into two groups for bolt stress analysis; namely, the self-energizing and the non-self-energizing gaskets. The self-energizing gasket is a gasket that generates a negligible axial gasket load and requires only an inconsequential amount of bolt force to produce an initial seal. The self-energizing gasket encompasses most of the sealing devices which are sometimes called seals. In Ref. 5 sealing devices are divided into two groups: seals and gaskets. A seal is defined as a device which is capable of providing dynamic sealing between two members which have relative motions, whereas a gasket is defined as a device for static sealing between two members which are clamped together. However, some of the devices such as an O ring seal can serve as both static and dynamic seals. Thus, an O ring can be called a seal or a gasket dependent of its application. To avoid confusion, all sealing devices are called gaskets in this report and they are classified only as self-energizing and non-self-energizing gaskets according to the preceding ASME definitions. As depicted in Fig. 2.1, many of the sealing devices used in shipping casks are O rings.

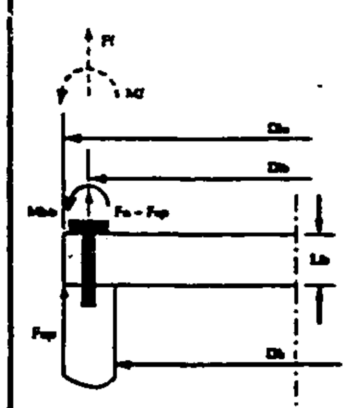
By definition, no gasket loads need to be considered in the bolt stress analysis for the self-energizing gaskets. However, for the non-self-energizing gaskets, two gasket loads must be considered; namely, an operating gasket load and a gasket seating load. A non-self-energizing gasket normally requires a high initial installation load to smooth out the roughness of the contact surfaces and to achieve a uniform compression in the gasket. Experience has shown that the gasket will not be leakproof unless such a seating operation is carried out and a minimum residual load is maintained on the gasket afterwards. Both the gasket seating load and the gasket operating load must be considered in the bolt stress analysis if they are supplied by the bolts. The gasket seating load can be much higher than the minimum operating gasket load and the design bolt preload.

Table 2.1 Formula for Evaluating Maximum Prying Tensile Bolt Force Generated by Applied Loads

Load Case	Figure	Formula for Bolt Force	Parameter Definition
Outward load applied in the direction normal to the closure lid. Its magnitude is represented by the fixed-edge force (Ff) and moment (Mf) that it generates at the bolt circle.		Additional tensile bolt force per bolt (Fap) caused by prying action of closure lid $Fap = \left(\frac{\pi Dlb}{Nb}\right)\left[\frac{\frac{2\,Mf}{(Dlo - Dlb)} - C1(B - Ff) - C2(B - P)}{C1 + C2}\right]$ where $C1 = 1$ $C2 = \left(\frac{8}{3\,(Dlo - Dlb)^2}\right)\left[\frac{El\,tl^3}{1 - Nul} + \frac{(Dlo - Dli)\,Elf\,tlf^3}{Dlb}\right]\left(\frac{Lb}{Nb\,Db^2\,Eb}\right)$ $B = Ff$ if $Ff > P$, otherwise $B = P$ B is the non-prying tensile bolt force, and F is the bolt preload. B, P, Ff and Mf are quantities per unit length of bolt circle. To convert a value per bolt to a value per unit length of bolt circle, multiply the value with the factor $[Nb/(\pi Dlb)]$.	Db: Nominal diameter of the closure bolt Dlb: Closure lid diameter at the bolt circle Dli: Closure lid diameter at the inner edge Dlo: Closure lid diameter at the outer edge Eb: Young's modulus of the closure bolt material El: Young's modulus of the closure lid material Elf: Young's modulus of the closure lid flange material Ib: Bolt area moment of inertia per unit length of the bolt circle Ff: Fixed-edge force of the closure lid at the bolt circle caused by the applied load (per unit length of the bolt circle) Lb: Bolt length between the top and bottom surfaces of the closure lid at the bolt circle Mf: Fixed-edge moment of the closure lid at the bolt circle caused by the applied load (per unit length of the bolt circle) Nb: Total number of closure bolts NUl: Poisson's ratio of the closure lid material P: Bolt preload per unit length of the bolt circle π: 3.1416 tl: Thickness of the closure lid tlf: Thickness of the flange of the closure lid

Notes: The listed formulas can be used with any consistent set of units for the parameters. Assumptions for the presented formula are as follows: rigid cask wall, flexible closure lid and bolt, and identical bolts equally spaced at bolt circle. See Appendix III for further details on the basis of the presented formulas. Formulas for Mf and Ff are given in individual load tables (Tables 4.1–4.8). The formulas for Fap are those given in Appendix III for the plate-ring model. The formulas of the plate-plate model may also be used.

Table 2.2 Formula for Evaluating Maximum Bending Bolt Moment Generated by Applied Loads

Load Case	Figure	Formula for Bolt Moment	Parameter Definition
Outward or inward load applied in the direction normal to the closure lid. Its magnitude is represented by the fixed-edge force (Ff) and moment (Mf) that it generates at the bolt circle.		Bending bolt moment per bolt (Mbb) caused by bent or rotated closure lid $Mbb = \left(\frac{\pi Dlb}{Nb}\right)\left(\frac{Kb}{Kb + Kl}\right) Mf$ where $Kb = \left(\frac{Nb}{Lb}\right)\left(\frac{Eb}{Dlb}\right)\left(\frac{Db^4}{64}\right)$ $Kl = \frac{El\,tl^3}{3\left[(1 - NUl^2) + (1 - NUl)^2\left(\frac{Dlb}{Dlo}\right)^2\right] Dlb}$	Db: Nominal diameter of the closure bolt Dlb: Closure lid diameter at the bolt circle Dli: Closure lid diameter at the inner edge Dlo: Closure lid diameter at the outer edge Eb: Young's modulus of the closure bolt material El: Young's modulus of the closure lid material Ff: Fixed-edge force of the closure lid at the bolt circle caused by the applied load (per unit length of the bolt circle) Lb: Bolt length between the top and bottom surfaces of the closure lid at the bolt circle Mf: Fixed-edge moment of the closure lid at the bolt circle caused by the applied load (per unit length of the bolt circle) Nb: Total number of closure bolts NUl: Poisson's ratio of the closure lid material π: 3.1416 tl: Thickness of the closure lid

Notes: The listed formulas can be used with any consistent set of units for the parameters. Assumptions for the presented formula are as follows: rigid cask wall, flexible closure lid and bolt, rigid joint between lid and bolt, and identical bolts equally spaced at bolt circle. See Appendix IV for further details on the basis of the presented formulas. Formulas for Mf and Ff are given in individual load tables (Tables 4.1–4.8). The formula for Mbb is developed in Appendix IV of this report.

2.4 Impact Protection for the Closure Lid and Bolt Head

Figure 2.1 shows the different types of designs for closure lids and bolt heads considered here. The unprotected lid and bolt head expose the closure bolt to a transverse shear load during a free drop, while the protected lid and bolt head are shielded from the same shear load. Thus, shear loads must be considered for the unprotected lid and bolt head in the bolt analysis.

2.5 Application of Preload and Possible Scatter of Preload

Most closure bolts used for shipping casks are preloaded using a torque wrench and a prescribed torque value which is specified in the cask operation procedure. Preloading using a torque generates a torsional bolt moment in addition to a tensile bolt force. This torsional moment may remains as a residual moment after the preload torque is removed. This residual bolt torque and the residual bolt preload may be lower than the applied torque and the intended or target preload because of stress relaxation.

Tests have shown that preloading using a torque wrench is an unreliable operation (although its reliability can be significantly improved with increasing lubrication). Past tests have shown that applying the same torque may produce preloads varying as much as 30% above and 30% below the target preload. The actual preload range should be experimentally determined. To obtain an accurate repeatable bolt preload clamping force in the joint a stud tensioning device should be used.

The knowledge of the actual preload range is needed not only for the assessment of the effectiveness of the preload and the gasket, but also for the bolt stress analysis. As discussed earlier and shown in Appendices I, III, and IV, the preload can have significant effects on the bolt force and bending moment. Moreover, the most significant effects occur when the bolted joint is about to open (i.e., when the applied load is about to exceed the preload). Accordingly, the maximum preload and the maximum applied load are not the only critical conditions for the bolt analysis—the combination of an applied load which equals the preload should also be considered.

The amount and variation of the preload force can be difficult to predict and control. The preload force depends on the materials of the bolts and closure joint including the heat treatment, the geometry of the joint, the type and clearance of the threads, surface finishes, the presence of nicks and burrs, and the use of platings and lubricants. In addition, as discussed in Section 2.3, the use of self-energized or non-self-energized gaskets can affect the required preload. Good engineering design practices try to eliminate or minimize friction and gasket loading effects on the joint to produce a reliable, repeatable clamping force in the joint. Good practices consider the use of bolting materials which differ from the closure materials to reduce friction and the possibility of gouging and seizing. However, the selection of the materials must also consider other differences in their properties such as thermal expansion which can affect the preload at high and low temperature conditions. Frequently, platings and lubricants are used to reduce friction and gouging, but their compatibility with the bolting and closure materials must be considered in their application. The type of clearance and surface finish of the threads must be carefully selected to assure that a good tight joint with repeated load application can be obtained. There should be no visible nicks or burrs present in the threaded parts which can affect their function. A quality assurance program as described in Section 8.0 must specify strict quality standards and controls to ensure that the bolted joint parts are properly procured and maintained throughout their useful life cycle.

3.0 LOADINGS FOR CLOSURE BOLT STRESS ANALYSIS

3.1 Bolt Loadings

Some of the loadings experienced by the closure bolt are directly related to the design and operation of the bolted joint. These loadings (which have been introduced in the previous section) are the gasket seating load, the gasket (operation or sealing) load, and the bolt preload. If the bolt preload is applied using a torque wrench, an applied torsional load will also be present during the preload operation and a residual torsional load will exist after the operation. To determine the bolt forces/moments, these bolt loadings must be considered together with the cask loadings described in the following subsection.

3.2 Cask Loadings

The cask loadings correspond to the normal and hypothetical accident transport conditions specified in Federal Regulation 10 CFR 71 (Ref. 5). To facilitate the presentation of the analysis method, these loadings are classified in this report according to their cause and analysis method in the following manner: pressure load, temperature load, impact load, puncture load, and the vibration load. The impact load refers to the free drop conditions specified in the federal regulation. The regulation specifies more than one load condition in each of these load categories. All of these specified load conditions must be considered for the bolt analysis.

Some shipping casks also have special pre-operation test requirements which may cause excessive loads on the closure bolt. These loads must also be identified and included as normal conditions for the analysis.

3.3 Load Combination

All concurrent loadings must be combined to form load cases for closure bolt analysis. To identify the most critical load case, the bolt stresses of all the possible load cases must be obtained and compared according to the criteria defined in Section 7 of this report. Because of the complex interaction among the loads and the bolt forces/moments (as discussed in Appendix I), the combination method of the bolt forces/moments varies with the load. This subject is further discussed in Section 4 of this report.

4.0 BOLT FORCES/MOMENTS FOR CLOSURE BOLT STRESS ANALYSIS

4.1 Bolt Forces/Moment Characteristics

Details of the nature, cause, and relative significance of the various bolt forces/moments appear in Appendix I. The bolt forces/moments to be considered in the bolt analysis may include some or all of the following bolt forces and moments: the axial tensile bolt force, the transverse shear bolt force, the bending bolt moment, and the torsional bolt moment.

The axial tensile bolt force is the primary force in the bolt—almost all loads and deformations can generate a tensile bolt force. The transverse shear bolt force is significant only if the closure lid, and the bolts are not protected from transverse movement. Significant bending bolt moment does not appear because the bolted joint is designed so as not to depend on the the bolt moment to support loads. A significant torsional bolt moment is generated only in preloading using a torque wrench.

For a typical bolt, Ref. 2 shows that approximately 50% of the preload torque applied at the bolt head is needed to overcome the friction between the bolt head and the closure lid. Only the remaining 50% of the torque is transmitted to the bolt body. Eighty percent of this transmitted torque, or 40% of the applied torque, is used to overcome the thread friction. Thus, only 10% of the applied torque is actually used to stretch the bolt body in order to generate the preload. Accordingly, it is reasonable to assume for the stress analysis of closure bolts that during a preload operation, the torsional bolt moment never exceeds 50% of the applied torque, and after the preload operation, the residual torsional bolt moment is between 10% and 50% of the preload torque.

The axial tensile bolt force has a non-prying and a prying component. The non-prying component is the basic tensile bolt force caused by the direct action of the load and can be obtained by simply considering the equilibrium of the bolt force and the applied load. The prying component of the tensile bolt force is an additional force which has an appreciable magnitude only under certain conditions. Similar to the bending bolt moment, the prying tensile bolt force is caused by the load-induced bending deformation of the closure lid and can only be obtained by considering both the equilibrium of forces and the compatibility of displacements among the interacting components of the bolted closure.

Appendices III and IV develop approximate and conservative formulas for the evaluation of the prying bolt force and the bending bolt moment. Using finite element models with increasing realism, the appendices also assess the accuracy of the approximate formulas and identify simple design rules to minimize the prying force and the bending moment. The results show that the simplified formulas do not have excessive conservatism and are, therefore, adequate for the bolt analysis. Furthermore, gross permanent deformations of the bolt are more likely to be caused by the tensile bolt force rather than the bending bolt moment. The bending bolt moment, however, can still have a significant role in producing incremental permanent deformation, fatigue, and other failures which are sensitive to local and peak stresses.

Appendix III also shows that the prying force can be generated by both inward and outward applied loads. An inward applied load is like an external pressure whose resultant force is directed towards the cask interior and an outward load is like an internal pressure whose resultant force is directed towards the cask exterior. In the case of an outward load, the maximum prying action occurs when the applied load is equal to the preload. In the case of the inward load, the maximum prying action occurs when there is no preload. The maximum prying bolt force can be higher than the non-prying bolt force; therefore, it must not be neglected in the bolt analysis.

The bolt forces/moments are further discussed in the following subsections for each of the loadings identified in Section 3.

4.2 Bolt Forces/Moments Generated by Preload

Table 4.1 identifies all of the significant bolt forces/moments generated by the preload operation employing a torque wrench. As discussed in the preceding subsection, the approximate formulas relating the applied torque to the tensile bolt force and torsional bolt moment are empirical relations obtained from Reference 2.

Table 4.1 using K values or nut factors shows a wide range of reported K values for the calculation of the tensile bolt force. This scatter of K values confirms the discussion in Section 2.5 concerning the possible scatter of actual preloads generated by a torque wrench. For bolt stress analysis, the entire range of preload values should be considered, and the preload that gives the most conservative analysis should be used.

If all of the closure bolts are preloaded following a proper sequence and applied in many small load increments to assure a nearly uniform and simultaneous tightening of all the bolts, appreciable bolt prying action should not appear unless the closure lid is initially warped. Therefore, Table 4.1 omits information for prying calculations.

In Table 4.1 the residual tensile bolt force (Far) and the residual torsional bolt moment (Mtr) are the same as the applied or target preload and torsional bolt moment. This result implies no relaxation of the bolt force and moment after the preload operation. The current information on the subject of preload relaxation is confusing and inconclusive. However, if significant relaxation of the preload is known to occur in the bolted closure to be analyzed, the maximum possible reduction should be identified in order to establish the range of preload values for the bolt analysis.

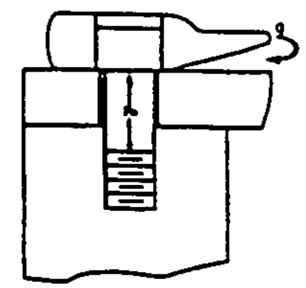
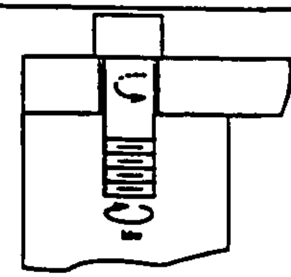
4.3 Bolt Forces/Moments Generated by Gasket Loads

Table 4.2 identifies all of the bolt forces/moments generated by the gasket seating load and the minimum gasket sealing or operation load.

The formulas for the tensile bolt force (Fa), are basically the empirical formulas given in ASME B&PVC, Section III, Appendix E for gasket loads (Ref. 3). The ASME formulas determine the gasket seating load and the minimum gasket sealing load using two empirical gasket factors or constants; namely, the *m* factor and the *y* factor. The *m* factor is the ratio of the required minimum gasket pressure to the pressure contained by the gasketed joint. The *y* factor is the minimum design seating stress of the gasket. The constants are experimentally determined. However, the experiments and results which established these constants were never published, and the values given in the ASME code for these empirical constants of various gaskets were simply presented as suggested values. Because of this uncertain beginning, the basis of the ASME formulas was not well understood and the validity of the formulas and the gasket factors have been questioned in the past. In recent years, the Pressure Vessel Research Council (PVRC) has sponsored a series of experimental studies aiming at reexamining the basis of the ASME formulas. The results of these studies have in essence confirmed the ASME approach to the characterization of gasket behavior. The study results have shown that the mechanical behavior of a gasket can be defined in terms of a few empirical constants. Moreover, it is possible to correlate these constants with the leak rate of the gasketed joint. The second edition of Ref. 2 has provided a summary of the findings of these studies and has suggested several possible ways to apply the findings to the design of leak-proof gasketed joints.

The formula for the torsional bolt moment generated by the gasket seating operation is based on the empirical relations between the applied torque and the tensile bolt force and between the applied torque and the torsional bolt moment given in Table 4.1.

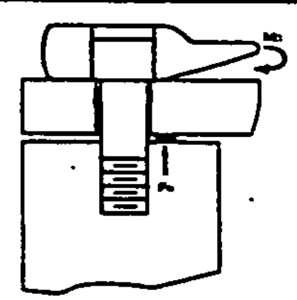
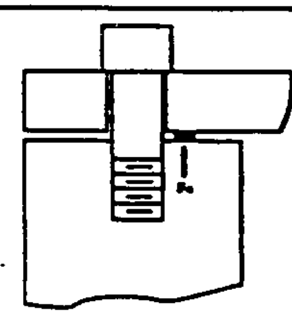
Table 4.1 Formulas for Evaluating Bolt Forces/Moments Generated by Preload

Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Applied preload using a torque wrench.		Non-prying tensile bolt force per bolt (Fa) $Fa = \frac{Q}{K\ Db}$ (same as the intended or target preload) Torsional bolt moment per bolt (Mt) $Mt = 0.5\ Q$ The applied preload does not have appreciable prying action, if the preload is applied in small increments and a proper sequence among all the bolts is followed.	Db: Nominal diameter (in.) of the closure bolt K: Nut factor for empirical relation between the applied torque and the achieved preload Q: The applied torque (in.-lb) for the preload Typical K values for steel fasteners, Bickford (Ref. 2, Ed. 1)
Residual stress after preload operation.		Maximum residual tensile bolt force (preload) per bolt (Far) $Far = Fa$ (same as the intended or target preload) Maximum residual torsional bolt moment per bolt (Mtr) $Mtr = 0.5\ Q$	

Lubricant	Reported Range	Reported Mean
As-received steel	0.158–0.267	0.1996
As-received cad plate	0.106–0.25	0.186
Fel-Pro C5A	0.08–0.23	0.132
Moly grease	0.1–0.16	0.137
Parkerized and oiled		0.177
Petroleum, light oils	0.099–0.15	0.123
Phosphate and oil	0.15–0.23	0.19

Notes: The listed formulas must be used with the units identified in the parameter definition column. Assumptions for the presented formulas are as follows: no lock nut is used; i.e., the applied torque (Q) does not include the "prevailing" torque required to run down a lock nut. About 50% of the applied torque (Q) is used to overcome friction between the bolt head and the closure lid, and no relaxation of residual bolt torsion. See Subsection 4.2 for further details on the basis of the presented formulas. The above typical K values were obtained from Ed. 1 of Ref. 2. The second edition of the same reference provides K values for steel fasteners with many other coatings or lubricants.

Table 4.2 Formulas for Evaluating Bolt Forces/Moments Generated by Gasket Loads

Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Gasket seating load using a torque wrench.		Non-prying tensile bolt force per bolt (Fa) generated by the gasket seating operation $Fa = \frac{\pi\ Dlg\ b\ y}{Nb}$ Torsional bolt moment per bolt (Mt) generated by the gasket seating operation $Mt = \frac{0.5\ \pi\ K\ Db\ Dlg\ b\ y}{Nb}$	b: Effective gasket or joint contact surface seating width (in.) as defined in ASME BPV Code, Section III, Appendix E Db: Nominal diameter (in.) of the closure bolt Dlg: Closure lid diameter (in.) at the location of the gasket load reaction, same as the parameter G defined in ASME BPV Code, Section III, Appendix E m: Gasket factor for operating conditions as given in ASME BPV Code, Section III, Appendix E Nb: Total number of closure bolts π: 3.1416 Pli: Pressure (psi) inside the closure lid Plo: Pressure (psi) outside the closure lid y: Minimum design seating stress (psi) of gasket as given in ASME BPV Code, Section III, Appendix E
Minimum operating gasket load (sufficient to maintain a tight joint).		Non-prying tensile bolt force per bolt (Fa) generated by the operating gasket load $Fa = \frac{2\ \pi\ Dlg\ b\ m\ (Pli - Plo)}{Nb}$ The prying action of gasket loads is minimal and neglected.	

Notes: The listed formulas must be used with the units identified in the parameter definition column. Assumptions for the presented formulas are as follows: rigid cask wall and closure lid and identical bolts uniformly spaced at bolt circle. See Subsection 4.3 for further details on the basis of the presented formulas. The ASME formulas for calculating the gasket seating load and the minimum operating gasket load are used here. The formulas are given in ASME BPV Code, Section III. Equivalent data from the gasket manufacturer may be used in lieu of the ASME formulas.

The gasket is usually located very close to the bolt circle. Thus, the gasket loads produce negligible moment about the bolt circle and, consequently, insignificant prying bolt force and bending bolt moment.

4.4 Bolt Forces/Moments Generated by Pressure Loads

Table 4.3 identifies all of the bolt forces/moments generated by an internal pressure load. The formula for the non-prying tensile bolt force is obtained by equating the sum of the tensile bolt forces of all of the bolts to the total net pressure load over the lid area bound by the gasket. The shear bolt force is obtained by equating the radial displacement of the closure lid to the radial displacement of the cask wall which is caused by internal pressure of the cask.

In addition to the non-prying tensile bolt force, the pressure load can also produce, by prying, an additional tensile bolt force and a bending bolt moment. The fixed-edge force (Ff) and moment (Mf) listed in Table 4.3 are to be inserted into the formulas listed in Tables 2.1 and 2.2 for the determination of the prying bolt force and the bending moment. The definition and equation for the calculation of Ff and Mf are given in Appendix III. The formulas listed in Table 4.3 for Ff and Mf are obtained by using the equations in Appendix III and simply assuming that pressure (P) covers the entire closure lid surface area within the bolt circle.

If the load is an external pressure, the non-prying tensile bolt force (Fa) will vanish because the load on the closure lid is supported by the cask wall and produces no axial force in the closure bolts. This result holds as long as the closure lid does not bend under the external pressure load. However, if it bends, the bending lid will cause a prying action on the closure bolts. As discussed in Appendices III and IV, the same formulas listed in Tables 2.1 and 2.2 can be used for determining the resulting prying bolt force and bending moment provided the variable substitution specified in Table 2.1 is implemented to accommodate the change of load direction from an inward to an outward load.

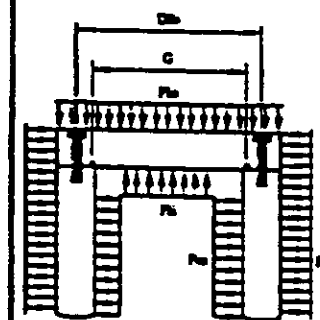
4.5 Bolt Forces/Moments Generated by Temperature Loads

A non-uniform thermal expansion in the bolted-joint and components can produce forces and moments in the closure bolts (i.e., temperature loads on the closure bolts). The non-uniform thermal expansion can be attributed to the difference in the temperatures or in the thermal-expansion coefficients of the joint components. Table 4.4 identifies three common cases of non-uniform thermal expansion which can produce appreciable temperature loads on the closure bolts. The table also identifies for each case the dominant bolt forces or moments generated and the approximate formulas for their calculation.

The temperature loads themselves may not be significant in the closure bolt because of the similarity of joint-component materials and the efficiency of heat transfer within and among the joint components. However, a temperature load is like a preload; any tensile bolt force that it produces is added to the existing tensile bolt preload which may be very high already. Frequently, in shipping casks, the initial or operating bolt preload is set for an accident condition. If this is the case, the extreme temperature of the fire accident will make the temperature load a critical condition to consider in the bolt analysis.

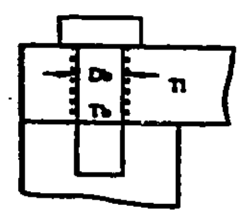
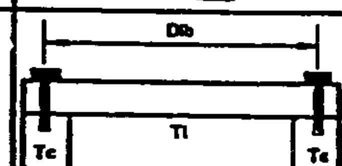
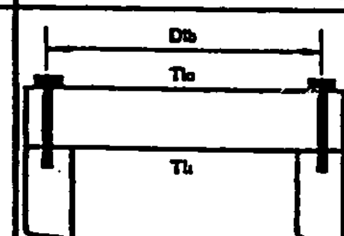
The formulas listed in Table 4.4 for the calculation of the non-prying tensile bolt force produced by the first temperature load case (the thermal-expansion difference between closure lid and the bolt) is based on the assumption that the lid is rigid and that the bolt force is required to produce a bolt extension equal to the difference of thermal expansions of the lid and the bolt. The assumption for

Table 4.3 Formulas for Evaluating Bolt Forces/Moments Generated by Pressure Loads

Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Load caused by the pressure difference between the interior and the exterior of closure components.		Non-prying tensile bolt force per bolt (Fa) $Fa = \dfrac{\pi\, Dlg^2\,(Pli - Plo)}{4\, Nb}$ Shear bolt force per bolt (Fs) $Fs = \dfrac{\pi\, El\, tl\,(Pci - Pco)\, Dlb^2}{2\, Nb\, Ec\, tc\,(1 - NUl)}$ Fixed-edge closure-lid force (Ff) and moment (Mf) to be inserted into the formulas listed in Tables 2.1 and 2.2 for the calculation of prying tensile bolt force (Fap) and bending bolt moment (Mbb) $Ff = \dfrac{Dlb\,(Pli - Plo)}{4}$ $Mf = \dfrac{(Pli - Plo)\, Dlb^2}{32}$	Dlb: Closure lid diameter at the bolt circle Dlg: Closure lid diameter at the location of gasket load reaction Ec: Young's modulus of the cask wall material El: Young's modulus of the closure lid material Nb: Total number of closure bolts NUl: Poisson's ratio of the closure lid material Pci: Pressure inside the cask wall Pco: Pressure outside the cask wall π: 3.1416 Pli: Pressure inside the closure lid Plo: Pressure outside the closure lid tc: Thickness of the cask wall tl: Thickness of the closure lid

Note: The listed formulas can be used with any consistent set of units for the parameters. Assumptions for the presented formulas are as follows: unbendable closure lid and cask wall and identical bolts equally spaced at bolt circle. See Subsection 4.4 for further details on the basis of the presented formulas. The formulas for Ff and Mf are obtained from Eqs. (III.44) and (III.45) of Appendix III with the diameter of the pressure area set to Dlb.

Table 4.4 Formulas for Evaluating Bolt Forces/Moments Generated by Temperature Loads

Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Load caused by thermal-expansion difference between the closure lid and bolt.		Non-prying tensile bolt force per bolt (Fa) $Fa = 0.25\,\pi\, Db^2\, Eb\,(al\, Tl - ab\, Tb)$	ab: Thermal expansion coefficient of the closure bolt material ac: Thermal expansion coefficient of the cask wall material al: Thermal expansion coefficient of the closure lid material Db: Nominal diameter of the closure bolt Dlb: Closure lid diameter at the bolt circle Eb: Young's modulus of the closure bolt material El: Young's modulus of the closure lid material Nb: Total number of closure bolts NUl: Poisson's ratio of the closure lid material π: 3.1416 Tb: Temperature change of the closure bolt Tc: Temperature change of the cask wall tlf: Thickness of the closure lid flange Tli: Temperature change of the inner surface of the closure lid Tlo: Temperature change of the outer surface of the closure lid Tl: Temperature change of the closure lid tl: Thickness of the closure lid
Load caused by thermal-expansion difference between the closure lid and cask wall.		Shear bolt force per bolt (Fs) $Fs = \dfrac{\pi\, El\, tlf\, Dlb\,(al\, Tl - ac\, Tc)}{Nb\,(1 - NUl)}$	
Load caused by temperature gradient between the inner and outer surfaces of the closure lid.		Fixed-edge closure-lid force (Ff) and moment (Mf) to be inserted into the formulas listed in Tables 2.1 and 2.2 for the calculation of prying tensile bolt force (Fap) and bending bolt moment (Mbb) $Ff = 0$ $Mf = \dfrac{El\, al\, tl^2\,(Tlo - Tli)}{12\,(1 - NUl)}$	

Note: The listed formulas can be used with any consistent set of units for the parameters. Assumptions for the presented formulas are as follows: rigid cask wall, rigid closure lid in thickness direction, and identical bolts equally spaced at bolt circle. See Subsection 4.5 for further details on the basis of the presented formulas. All temperature changes are measured from the stress-free temperature. The thermal expansion coefficient is an average value for the temperature range. No prying and bending are generated by the first two load cases, while the third (temperature-gradient) load case produces only a prying and bending action. The formulas for Ff and Mf of this case are the same as Eqs. (III.46) and (III.47) given in Appendix III.

Table 4.5 Formulas for Evaluating Bolt Forces/Moments Generated by Impact Load Applied to a Protected Closure Lid

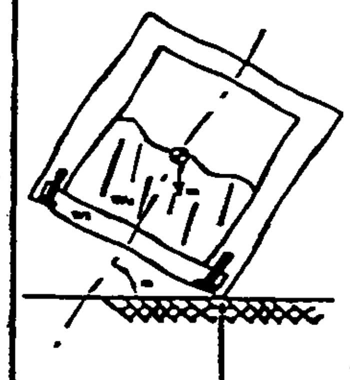
Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Load caused by impact for a cask with a protected closure lid.		Non-prying tensile bolt force per bolt (Fa) $Fa = \frac{1.34 \sin(xi)\ DLF\ ai\ (Wl + Wc)}{Nb}$ Shear bolt force per bolt (Fs) $Fs = \frac{\cos(xi)\ ai\ Wl}{Nb}$ Fixed-edge closure-lid force (Ff) and moment (Mf) to be inserted into the formulas listed in Tables 2.1 and 2.2 for the calculation of prying tensile bolt force (Fap) and bending bolt moment (Mtb) $Ff = \frac{1.34 \sin(xi)\ DLF\ ai\ (Wl + Wc)}{\pi\ Dlb}$ $Mf = \frac{1.34 \sin(xi)\ DLF\ ai\ (Wl + Wc)}{8\pi}$	ai: Maximum rigid-body impact acceleration (g) of the cask Dlb: Closure lid diameter at the bolt circle DLF: Dynamic load factor to account for any difference between the rigid body acceleration (ai) and the acceleration of the contents and closure lid π: 3.1416 Nb: Total number of closure bolts Wc: Weight of the cask contents Wl: Weight of the closure lid xi: Impact angle between the cask axis and the target surface
Note: The listed formulas can be used with any consistent set of units for the parameters, except the impact acceleration which must be measured in g. Assumptions for the presented formulas are as follows: rigid closure lid and cask wall and identical bolts equally spaced at bolt circle. See Subsection 4.6 for further details on the basis of the presented formulas.			

Table 4.6 Formulas for Evaluating Bolt Forces/Moments Generated by Impact Load Applied to an Unprotected Closure Lid

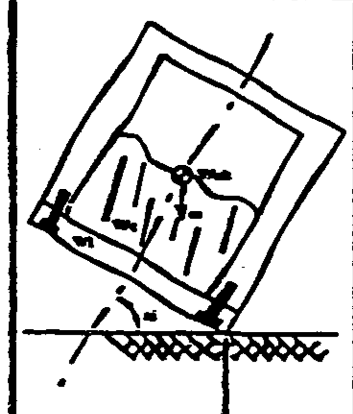
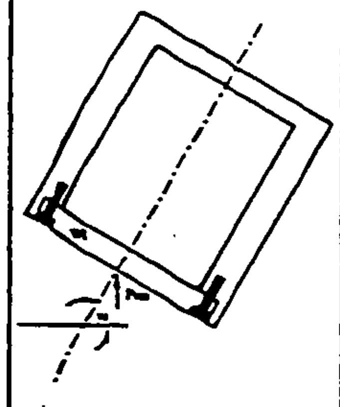
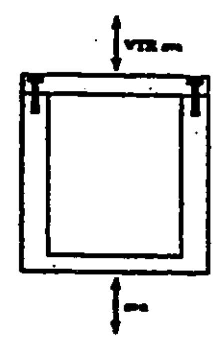
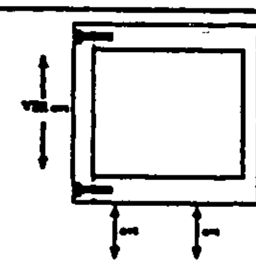
Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Load caused by impact for a cask with an unprotected closure lid.		Non-prying tensile bolt force per bolt (Fa) $Fa = \frac{1.34 \sin(xi)\ DLF\ ai\ (Wl + Wc)}{Nb}$ Shear bolt force per bolt (Fs) $Fs = \frac{\cos(xi)\ ai\ Wck}{Nb}$ Note: For side impact with identical impact limiters at the cask ends, only one-half of the total cask weight is needed. Fixed-edge closure-lid force (Ff) and moment (Mf) to be inserted into the formulas listed in Tables 2.1 and 2.2 for the calculation of prying tensile bolt force (Fap) and bending bolt moment (Mtb) $Ff = \frac{1.34 \sin(xi)\ DLF\ ai\ (Wl + Wc)}{\pi\ Dlb}$ $Mf = \frac{1.34 \sin(xi)\ DLF\ ai\ (Wl + Wc)}{8\pi}$	ai: Maximum rigid-body impact acceleration (g) of the cask Dlb: Closure lid diameter at the bolt circle DLF: Dynamic load factor to account for any difference between the rigid body acceleration (ai) and the acceleration of the contents and closure lid π: 3.1416 Nb: Total number of closure bolts Wck: Total weight of the cask Wc: Weight of the cask contents Wl: Weight for the closure lid xi: Impact angle between the cask axis and the target surface
Note: The listed formulas can be used with any consistent set of units for the parameters, except the impact acceleration which must be measured in g. Assumptions for the presented formulas are as follows: rigid closure lid and cask wall and identical bolts equally spaced at bolt circle. See Subsection 4.6 for further details, on the basis for the presented formulas.			

Table 4.7 Formulas for Evaluating Bolt Forces/Moments Generated by Puncture Loads

Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Load caused by puncture.		Non-prying tensile bolt force per bolt (Fa) $Fa = \frac{-\sin(xi)\ Pun}{Nb}$ Shear bolt force per bolt (Fs) $Fs = \frac{\cos(xi)\ Pun}{Nb}$ Fixed-edge closure-lid force (Ff) and moment (Mf) to be inserted into the formulas listed in Tables 2.1 and 2.2 for the calculation of prying tensile bolt force (Fap) and bending bolt moment (Mbb) $Ff = \frac{-\sin(xi)\ Pun}{\pi\ Dlb}$ $Mf = \frac{-\sin(xi)\ Pun}{4\ \pi}$ A minus sign is assigned to Fa, Ff, and Mf, because the puncture load is an inward load which is directed toward the cask interior.	Dlb: Closure lid diameter at the bolt circle Dpb: Puncture bar diameter Nb: Total number of closure bolts π: 3.1416 Pun: Maximum puncture load generated by the puncture bar Syl: Yield strength of the closure lid material Sul: Tensile strength of the closure lid material tl: Thickness of the closure lid xi: Impact angle between the cask axis and the target surface Pun is the maximum impact force that can be generated by the puncture bar during a normal impact. Appendix VI provides the derivation of the following formulas: Pun = The smaller of $(0.75\ \pi\ Dpb^2\ Syl)$ and $(0.6\ \pi\ Dpb\ tl\ Sul)$

Notes: The listed formulas can be used with any consistent set of units for the parameters. Assumptions for the presented formulas are as follows: rigid closure lid and cask wall and identical bolts equally spaced at bolt circle. Puncture load is determined by the penetration and indentation resistances of the closure lid. See Subsection 4.7 for further details on the basis of the presented formulas.

Table 4.8 Formulas for Evaluating Bolt Forces/Moments Generated by Vibration Loads

Load Case	Figure	Formulas for Bolt Forces/Moments	Parameter Definition
Vibration in the direction of the cask axis.		Tensile bolt force per bolt (Fa) $Fa = \frac{VTR\ ava\ Wl}{Nb}$ Fixed-edge closure-lid force (Ff) and moment (Mf) to be inserted into the formulas listed in Tables 2.1 and 2.2 for the calculation of prying tensile bolt force (Fap) and bending bolt moment (Mbb) $Mf = \frac{-\sin(xi)\ Pun}{8\ \pi}$	ava: Maximum axial vibration acceleration (g) at the cask support. For the analysis of tensile bolt force, ava is considered positive if it is directed toward the cask exterior. avt: Maximum transverse vibration acceleration (g) at the cask support π: 3.1416 Nb: Total number of closure bolts VTR: Vibration transmissibility of acceleration between the cask support and the closure lid Wl: Weight of the closure lid
Vibration in the direction normal to the cask axis.		Shear bolt force per bolt (Fs) $Fs = \frac{VTR\ avt\ Wl}{Nb}$	

Notes: The listed formulas can be used with any consistent set of units for the parameters, except the impact acceleration which must be measured in g. Assumptions for the presented formulas are as follows: rigid cask wall and closure lid and identical bolts equally spaced at bolt circle. See Subsection 4.8 for further details on the basis of the presented formulas. Vibration loads are insignificant unless a resonance condition exists or there is an excessive bending or prying action. See Subsection 4.8 for details. Axial vibration loads are both inward and outward loads. The formulas of Fa, Ff, and Mf given here are for both loads, provided ava is assigned the proper sign, i.e., the + sign for an outward acceleration and the sign for an inward acceleration.

the calculation of the shear bolt force of the second load case (the thermal-expansion difference between the closure lid and the cask wall) is that only the closure lid is deformed by the shear bolt force. Thus, the magnitude of the shear bolt force is determined by the condition that the radial displacement of the closure lid generated by shear bolt force at the bolt circle is equal to the difference of thermal expansions of the closure lid and the cask wall. This condition ensures that the calculated value of the shear bolt force is conservative for design purposes.

The third load case listed in Table 4.4 (the temperature gradient between the inner and outer surfaces of the closure lid) generates only a prying action. Therefore, only the formulas for the fixed-edge force (Ff) and moment (Mf) are given for the analysis of the prying effects. The formulas are obtained (Ref. 6) based on the fact that a linear temperature gradient through the thickness of a thin plate produces a uniform, pure bending of the plate. It should be noted that similar to the pressure load, the temperature gradient load also has a direction. The closure lid always bends towards the surface with the higher thermal expansion or temperature. Thus, the load can be an inward or an outward load relative to the cask interior. In both cases, a tensile prying bolt force can be produced and the force can be evaluated using the same Ff and Mf formulas given in Table 4.4 but using the appropriate formula in Tables 2.1 and 2.2 for inward and outward loads.

4.6 Bolt Forces/Moments Generated by Impact Loads

The formulas for calculating bolt forces/moments for impact loads are listed in Table 4.5 for a protected bolted closure and in Table 4.6 for an unprotected closure. The only difference between the two tables is in the magnitude of the shear bolt force. In the case of an unprotected lid, the lid receives the impact or inertial force of the entire cask while in the case of the protected lid, the lid feels only its own impact force. To derive the formulas shown in the tables, the impact force is obtained by multiplying the impacting mass with the peak impact acceleration of the shipping cask. The impact force is then decomposed into two components in the axial and the transverse directions of the cask. The axial force component provides the non-prying tensile bolt force, while the transverse component generates the shear bolt force. In obtaining the shear bolt force, the friction of the bolted joint between the closure lid and the cask wall is conservatively omitted. The main reason for this omission is the uncertainty concerning the coefficient of friction.

The distribution of the impact force to individual bolts is based on the assumption that the impact force produces a rigid body motion of the closure lid which in turn generates bolt forces that are proportional to the rigid displacement at the bolt locations. Assuming that the rigid-body motion of the closure lid in the transverse direction of the cask is a simple translation, the transverse impact force component is uniformly distributed to all the bolts to obtain the shear bolt forces given in Tables 4.5 and 4.6. Similarly, assuming that the motion of the lid in the axial direction of the cask is a simple rotation about the impact point, the axial impact force component is linearly distributed to all of the bolts. Thus, the bolt closest to the impact point receives the smallest tensile force and the bolt that is farthest from the impact point receives the largest force. The average bolt force is equal to the axial impact force divided by the number of bolts.

The non-prying tensile bolt forces listed in Tables 4.5 and 4.6 are the largest bolt forces. Mathematical analysis in Appendix V proves that, regardless of the impact angle and the lid diameter, the largest bolt force is always 1.34 times that of the average bolt force.

Similar to the pressure load, the axial impact load can also produce a prying action on the closure bolts. The fixed-edge force (Ff) and moment (Mf) given in Tables 4.5 and 4.6 for the prying analysis result from replacing the axial impact load with an equivalent pressure load whose magnitude is sufficient to produce the above-mentioned largest tensile bolt force of the impact load.

Table 4.9 Methods for Combining Bolt Forces from Different Loads

TENSILE BOLT FORCE

The tensile bolt forces must be carried out in the following steps to correctly include the complex interactions among the preload, the temperature loads, and the mechanical loads and between the non-prying and prying tensile bolt forces:

I. Combination of Non-prying Tensile Bolt Forces

(I.1) Use formulas in Tables 4.3–4.8 to calculate the non-prying tensile bolt force (Fa) generated by each of the loads to be combined, including the preload. Do not drop the sign of the bolt force. The + and – signs indicate the bolt force to be added to or subtracted from existing tensile bolt force.

(I.2) Sum up the tensile bolt forces obtained in Step I.1 for the operating preload and temperature load; identify the combined bolt force as Fa_pt.

(I.3) Sum up the tensile bolt forces obtained in Step I.1 for the remainder of the loads to be combined; identify the combined bolt force as Fa_al.

(I.4) Compare Fa_pt and Fa_al; use the larger of the two forces as the combined non-prying tensile bolt force. Identify the combined bolt force as Fa_c. Set Fa_c to zero, if it is less than zero.

II. Combination of Prying Tensile Bolt Forces

(II.1) Use formulas in Tables 4.3–4.8 to calculate the fixed-edge force (Ff) and moment (Mf) generated by each of the loads to be combined, including the temperature loads. Use + sign for the result of an outward load (directed towards the cask exterior) and – sign for an inward load.

(II.2) Sum up the forces and moments obtained in Step II.1 for all the loads to be combined including the temperature load. Identify the combined force and moment as Ff_c and Mf_c. If Mf_c is positive, the combined load is an outward load, otherwise it is an inward load.

(II.3) Use formulas in Table 2.1 to obtain the prying tensile bolt force for the combined load. Use Fa_pt to calculate the preload (P) required by the formulas. Identify the prying bolt force obtained for the combined load as Fab_c. If the combined load is an inward load, follow the instructions given in Table 2.1 for application of the formulas to inward loads.

III. Combination of the Combined Non-prying and Prying Tensile Bolt Forces

(III.1) Add the Fa_c obtained in Step I.4 and the Fap_c obtained in Step II.3 to obtain the total tensile bolt force for stress analysis.

SHEAR BOLT FORCE

The shear bolt force is evaluated only for unprotected bolt and closure lids that depend on the bolt to resist transverse shear load. The combined shear force (Fs_c) is the absolute sum of the shear forces (Fs) generated by all applied loads to be combined including the temperature loads. Use formulas in Tables 4.3–4.8 for the calculation of Fs.

BENDING BOLT MOMENT

The bending bolt moment and stress is not likely to cause large plastic bolt deformation, but it can cause incremental plastic bolt deformation and fatigue under cyclic loadings. To obtain the combined bending moment (Mbb_c), use the Mf_c obtained in Step II.2 above for the combined prying tensile bolt force and the formula in Table 2.2 for Mbb.

TORSIONAL BOLT MOMENT

Torsional bolt moment is generated only by the preload. No combination is needed.

In Tables 4.5 and 4.6, the cask is shown to impact at its top where the closure lid is located. This case is more critical than when the impact occurs at the cask bottom. In the case of bottom impact, the non-prying tensile bolt force vanishes—the shear force depends only on the lid mass, not the contents mass, and the axial impact force is an inward force.

In Tables 4.5 and 4.6, a dynamic load factor (DLF) is included in the formulas for the tensile bolt force in order to account for possible dynamic amplification of the cask rigid-body impact acceleration (ai). The amplification can be caused by the vibration response of the closure lid in the cask axial direction. Theoretically, the maximum possible value of this factor is 2.0 (Ref. 7).

4.7 Bolt Forces/Moments Generated by Puncture Loads

Although the impact energy of the entire cask is available to the puncture process, the puncture force is limited by the indentation and puncture resistances of the closure lid. The resistances are limited because they are determined by the impact area and the lid material strength (both of which have an upper limit). The formula given in Table 4.7 for the maximum puncture force (PUN) is in Appendix VI. Two possible upper limits of the puncture force are obtained in Appendix VI using two failure or deformation models of the closure lid. The formula in Table 4.7 for PUN simply states that the smaller of these two upper limits is used as the maximum puncture force for closure bolt stress analysis.

The formula gives PUN for an impact normal to the closure lid surface. For impact at an oblique angle, the same force applies but the force is broken into two components in the axial and transverse directions of the cask. The transverse force is divided equally among all of the bolts to provide the shear bolt force given in Table 4.7. The axial force does not transmit to the bolts because the puncture force is an inward load. It can produce a tensile bolt force only by prying. The prying tensile bolt force and bending bolt moment can be obtained using the fixed-edge force (Ff) and moment (Mf) given in Table 4.7 and the formulas in Tables 2.1 and 2.2 for an inward load. The formulas for Ff and Mf are obtained from Equations III.42 and III.43 in Appendix III which work for a concentrated load applied at the closure lid center.

4.8 Bolt Forces/Moments Generated by Vibration Loads

The vibration load is normally not significant unless a resonance condition exists or excessive prying/bending action is present. Appendix I shows that the non-prying tensile bolt force of a vibration load can always be effectively "masked" by a sufficiently large bolt preload but the same cannot be said of the prying bolt force. Thus, in analyzing bolt forces/moments generated by the vibration load, the attention should be focused on the possibility of resonance and prying.

For the formulas in Table 4.8, the possible effect of resonance is included in the vibration transmissibility of peak acceleration (VTR). The VTR relates the amplitude of the peak input vibration excitation to the peak structural acceleration response (Refs. 8 and 9). Theoretically, at a resonant frequency of the structure, the value of the VTR can go to infinity and is limited only by the damping of the structure. For conservatism, the minimum VTR value allowed for the formulas in Table 4.8 is 1.0.

To derive the formulas given in Table 4.8, the vibration load is conservatively treated as a uniform inertial load having an acceleration equal to the peak vibration acceleration. This inertial load is distributed equally to all of the bolts to obtain the non-prying tensile bolt force and the shear bolt force. The formulas for the fixed-edge force (Ff) and moment (Mf) are obtained from Equations III.44 and III.45 in Appendix III which are for a uniformly distributed load on the closure lid.

The vibration load must be considered both as an inward load and as an outward load in prying analysis. However, its effect is likely to be more significant as an outward load because an outward vibration load produces both non-prying and prying tensile bolt forces. The same Ff and Mf formulas work for both analyses.

4.9 Combination of Bolt Forces/Moments from Different Loads

As shown in Appendix I, the bolt forces and moments (especially the axial tensile force) are the result of a complex structural interaction among the various joint components and loadings. To make the solution feasible, the significant actions of individual loadings and bolt force/moment components are isolated and analyzed in the foregoing sections (Sections 4.2–4.8) using simplified assumptions. These results must be combined properly in order to correctly simulate the underlying phenomena that determine the actual bolt forces and moments.

Table 4.9 outlines the proper procedure for combining the bolt force/moment results from the various loads. In the evaluation of the tensile bolt force, this procedure takes into account the significant interactions between the preload and the temperature loads, between the preload and the applied loads, and between the non-prying and the prying bolt force components. The interaction between the tensile bolt force and the shear bolt force is, however, omitted in the evaluation of the shear bolt force mainly because of the uncertainty regarding the coefficient of friction. The interaction between the bending bolt moment and the prying tensile bolt force is also neglected because the finite element analyses in Appendix III shows that the effect of the interaction on the bolt stress analysis result is inconsequential.

5.0 CALCULATION OF BOLT STRESSES

The stresses generated by the bolt forces/moments in the bolt are identical to those of a simple beam with a solid, circular, cross-section. The standard beam formulas (Ref. 6) are given in Table 5.1 for the calculation of the average tensile and shear stresses over the bolt cross-section and of the maximum bending and torsional shear stresses at the circumference of the bolt cross-section. A formula is also provided to obtain the maximum stress intensity from these stresses which is defined in the ASME B&PVC, Section III, (Ref. 3) the difference between the maximum and minimum principal stresses.

The bolt cross-section used for stress calculation depends on whether the shank or the thread section of the bolt carries the load. However, for the axial bolt force, it is always the thread section. Effective bolt diameters to be used for stress calculations are given in Table 5.1. All of these diameters are determined by the nominal bolt diameter and the bolt thread pitch (both of which are explicitly specified in the bolt designation). Table 5.2a & b explains the contents of two frequently used designations.

Table 5.1 Formulas for Bolt Stress Evaluation

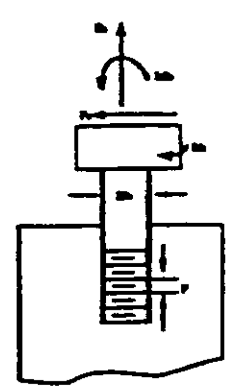
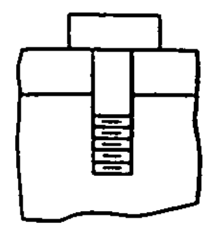
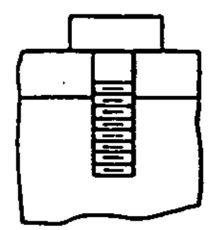
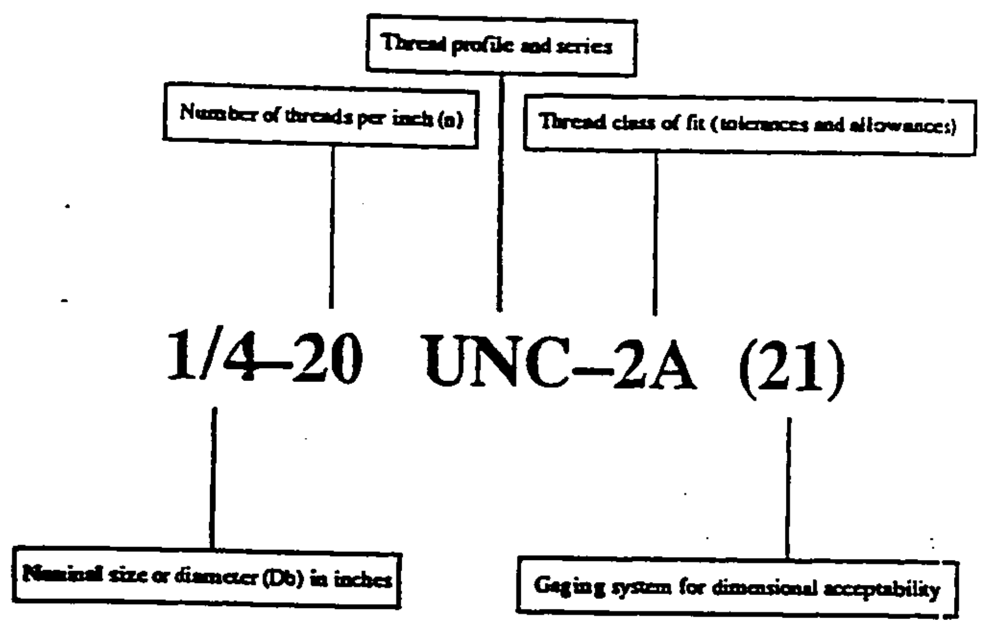
<table>
<tr><th colspan="2">Bolt Geometry, Forces/Moments and Stresses</th><th>Bolt Diameter To Be Used for Stress Calculation</th><th>Formulas for Bolt Stress Calculation</th></tr>
<tr><td rowspan="2"></td><td>Maximum shear and bending in the shank </td><td rowspan="2">n: Number of bolt threads per unit length
p: Bolt thread pitch, equal to 1/n
Db: Nominal diameter of the closure bolt
Dba: Bolt diameter for tensile stress calculation
= Db−0.9743 p   for inch-series threads
= Db−0.9382 p   for metric-series threads
Dbs: Bolt diameter for shear stress calculation
= Dba   if maximum shear occurs in the thread
= Db   if maximum shear occurs in the shank
Dbb: Bolt diameter for bending stress calculation
= Dba   if maximum bending occurs in the thread
= Db   if maximum bending occurs in the shank
Dbt: Bolt diameter for torsional stress calculation
= Dba</td><td rowspan="2">D: Bolt diameter used for stress calculation
Sba: Average tensile stress caused by the tensile bolt force (Fa)
= 1.2732 Fa / D^2
Sbs: Average shear stress caused by the shear bolt force (Fs)
= 1.2732 Fs / D^2
Sbb: Maximum bending stress caused by the bending bolt moment (Mb)
= 10.186 Mbb / D^3
Sbt: Maximum shear stress caused by the torsional bolt moment (Mt)
= 5.093 Mt / D^3
Sbi: Maximum stress intensity caused by tension + shear + bending + torsion
= [(Sba + Sbb)^2 + 4 (Sbs + Sbt)^2]$^{0.5}$</td></tr>
<tr><td>Maximum shear and bending in the thread </td></tr>
<tr><td colspan="4">Notes: The listed formulas, except those identified for a specific unit system, can be used with any consistent set of units for the parameters. If the bolt has a special profile with varying diameter along its length, the smallest diameter should be used for calculating the tensile stress.</td></tr>
</table>

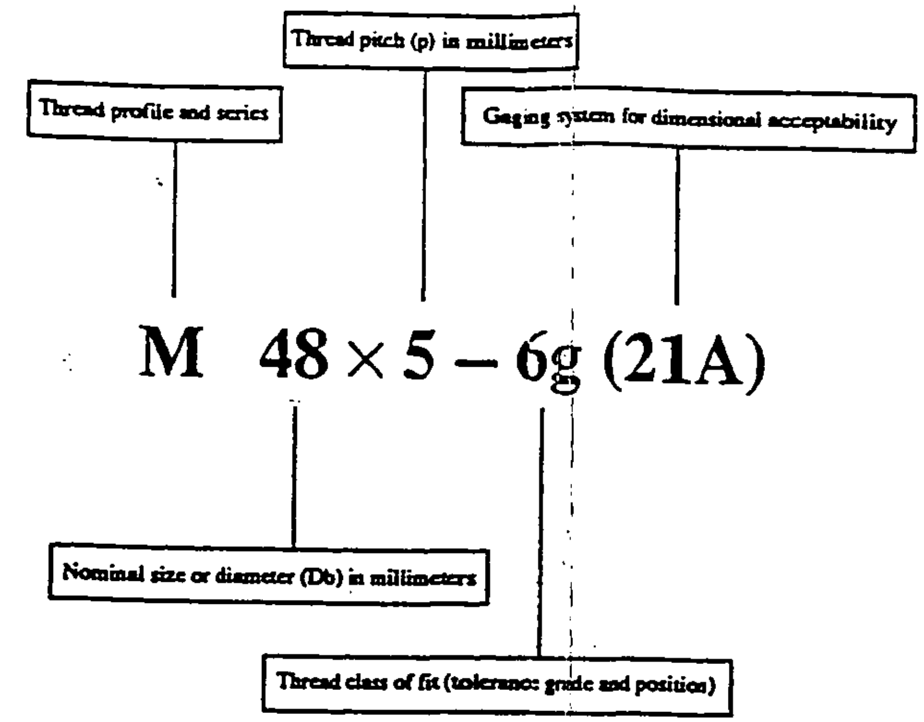
Table 5.2a Sample Bolt Thread Designations

Table 5.2b Sample Bolt Thread Designations

Unified inch-series threads:



Metric-series threads:



Standard Thread Profile

- UN Unified profile for inch-series threads with flat or rounded root for external (bolt) threads
- UNR UN profile with mandatory rounded-root radius of 0.108 to 0.144 times thread pitch
- UNJ UNR profile with larger root radius of 0.150 to 0.180 times thread pitch
  Optimal thread form for tensile and fatigue strengths, used for aerospace and other critical applications
- M Basic ISO 68 profile for metric-series threads with flat or rounded root for bolt threads, having same geometry as the UN profile
- MJ M profile corresponding to the UNJ profile

Common Thread Series (groups of diameter-pitch combinations)

- UN Unified constant-pitch series (for all diameters)
- UNC Unified coarse-pitch series
- UNF Unified fine-pitch series
- M Metric coarse-pitch series

6.0 CLOSURE BOLT STRESS ANALYSIS

6.1 Analysis Requirements and Criteria

Tables 6.1, 6.2, and 6.3 specify three sets of stress analysis requirements and the corresponding stress limits and acceptance criteria. The three sets of analyses specified are as follows:

(1) The maximum stress analysis of normal conditions.

(2) Fatigue stress analysis of normal conditions.

(3) Maximum stress analysis of accident conditions.

The analysis conditions refer to the normal and hypothetical accident conditions specified in Federal Regulation 10 CFR 71 (Ref. 5). Special loadings (such as the gasket seating load) that are unique to the closure bolt design and operation are considered as part of the normal conditions.

The specified analyses and acceptance criteria are based on ASME B&PVC, Section III, Subsection NB, for Class 1 nuclear power plant components (Ref. 3) with appropriate adjustments for the shipping cask. The requirements for the normal and accident conditions correspond to the ASME requirements for the Level A service and the Level D accident load conditions, respectively. To facilitate comparison, a summary of the ASME analysis and criteria requirements for these two load conditions appears in Appendix II of this report. For each of the requirements listed therein, the appendix also gives the identification number of the code subsection detailing that requirement.

6.2 Material Toughness Requirements

The specified acceptance criteria for closure bolt stress analysis assume a ductile bolt behavior. To assure that the bolt material has the required ductility, the material must meet the ASME Subsection NB requirements for bolting material testing and examination (Ref. 3). These requirements are summarized in Table 6.4.

A ductile material behavior is also implicitly required by the bolt force/moment analysis methods presented in this report. As explained in Appendix I, the methods intended for the analysis of the average behavior of the bolts cannot be used to accurately predict the results of individual bolts unless some stress redistribution is allowed to occur through local plastic deformation. In other words, the present analysis methods are not adequate for the stress analysis for bolted closures made of brittle material.

6.3 Basis for Stress Limits

The stress limits specified in Tables 6.1–6.3 for the closure bolt are more stringent than for the cask. This is the result of the recognition of the following differences between the cask and the closure bolt:

(1) The bolt material usually has much less ductility and work-hardening capacity than the cask material. The ultimate tensile strength and strain of a bolt material (especially one with high strength) are not much higher than the yield stress and strain.

(2) The structural behavior of the bolted joints is more complicated than that of the shipping cask. The stresses of individual bolts depend on many factors which cannot be precisely controlled or analyzed.

Table 6.1 Stress Analysis of Closure Bolts—Normal Conditions, Part I, Maximum Stress Analysis

Load Cases To Be Considered	Limits on Bolt Stresses Obtained Using Elastic Analysis
Gasket-seating load or the maximum applied preload	Sy: Minimum yield stress or strength of the bolt material Sm: Basic allowable stress limit for the bolt material, equal to 2/3 of Sy at the room temperature or 2/3 of Sy at the operating temperature, whichever is less.
Load combinations of all normal condition loads plus the minimum gasket load: Operating preload Minimum gasket load Pressure load Temperature load Impact load Vibration load	All of the following limits must be met: Tension Average stress < Sm (Allowable stress) Shear Average stress < 0.6 Sm (Allowable stress) Tension plus shear Stress ratio = computed average stress/allowable average stress Rt: Stress ratio for average tensile stress Rs: Stress ratio for average shear stress $Rt^2 + Rs^2 < 1$ Tension plus shear plus bending plus residual torsion For bolts having minimum tensile strength (Su) less than 100 ksi Maximum stress intensity < 1.5 Sm For bolts having minimum tensile strength (Su) greater than 100 ksi Maximum stress intensity < 1.35 Sm
Notes: The effect of prying, bending and residual torsional shear should be included. None of the normal loads are expected to govern the bolt design. The maximum applied preload is usually the worst load. See Subsection 6.5 for additional information.	Notes: See Subsection 6.3 for the basis of the stress limits. In the absence of bending and residual torsion, the tensile and shear stresses are governed by the limits on the average stresses. The limit for the combined stress condition is less restrictive, unless all stresses are present.

Table 6.2 Stress Analysis of Closure Bolts—Normal Conditions, Part II, Fatigue Stress Analysis

Load Histories To Be Considered	Acceptance Criteria
Repeated applied preload Load combinations of all normal condition loads plus the minimum gasket load: Operation preload Minimum gasket load Pressure load Temperature load Impact load Vibration load	Maximum cumulative usage factor (U) due to alternating stress intensity < 1.0 For bolts having minimum yield strength less than 100 ksi Use ASME Code, Section III, Appendix I, fatigue curves I-9.0 with elastic-modulus adjustment Use fatigue strength reduction factor not less than 4.0, unless it can be shown otherwise For bolts with minimum yield strength greater than 100 ksi Use ASME Code, Section III, Appendix I, fatigue curves I-9.4 with elastic-modulus adjustment Use fatigue strength reduction factor not less than 4 Thread shall be Vre-type having minimum root radius no less than 0.003 in. Fillet radius at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060 in.
Notes: The effect of prying, bending and residual torsional shear should be included. The repeated preload is usually the worst load, and it should be used to determine the allowable life of the closure bolt. The vibration load is not expected to be significant unless a resonance condition exists or excessive bending and prying action are present. Modify the design to eliminate these conditions. See Subsection 6.5 for additional information.	Notes: The specified fatigue curves are given in ASME BPV Code, Section III, Appendix I (Ref. 3).

Table 6.3 Stress Analysis of Closure Bolts—Accident Conditions, Maximum Stress Analysis

Loads To Be Considered	Limits on Bolt Stresses Obtained Using Elastic Analysis
	Sy: Minimum yield stress or strength of the bolt material Su: Minimum ultimate stress or strength of the bolt material
All accident condition loads: Impact Puncture Fire (temperature and pressure) Submersion (pressure)	All of the following limits must be met: Tension Average stress < The smaller of 0.7 Su or Sy at temperature (Allowable stress) Shear Average stress < The smaller of 0.42 Su or 0.6 Sy at temperature (Allowable stress) Tension plus shear Stress ratio = computed average stress/allowable average stress Rt: Stress ratio for average tensile stress Rs: Stress ratio for average shear stress $R_t^2 + R_s^2 \le 1$
Notes: The effect of the operating preload, bending and prying should be included. The impact load and the combined preload and fire load are expected to govern the bolt design. See Subsection 6.5 for additional information.	Notes: See Subsection 6.3 for the basis of the stress limits. The limit for the combined stress condition is less restrictive, unless all stresses are present.

Table 6.4 ASME Section III Requirements for Bolting Material of Class 1 Components

Requirement Category	Requirements	Code Section for Details
General	Bolt & stud material: Meet specification no. listed in Appendix I, Table I-1.3 Nut material: Meet specification no. listed in Appendix I, Table I-1.3 or SA-194 Washer material: Made of wrought material	NB2128
Fracture toughness	All bolting material including bolt, stud, and nut: Bolt, stud, and nut of nominal size > 1.0 in. shall be impact tested using the Charpy V-notch (Cv) method. Specimens from the bolting material shall be oriented in the axial direction and the notch normal to the surface. Three specimens shall be tested at the lower of the preload temperature or the lowest service temperature. All three specimens shall meet the following Cv requirement: Nominal diameter / Lateral expansion, mils / Absorbed energy, ft-lb 1 in. or less / No test required / No test required Over 1 in. to 4 in., incl. / 25 / No requirement Over 4 in. / 25 / 45 One test shall be made for each lot of material.	NB2311(a) NB2322.2 (a) NB2333 NB2345
Examination	Nominal size / Required examinations 1 in. or less / Visual in accordance with NB2582 Over 1 in. to 2 in., incl. / Visual plus the magnetic particle or the liquid penetrant Over 2 in. to 4 in., incl. / Visual plus the magnetic particle or the liquid penetrant, plus the ultrasonic as required by NB2585 Over 4 in. / Visual plus the magnetic particle or the liquid penetrant, plus the ultrasonic as required by NB2586	NB2580

The specified analyses (the maximum stress and fatigue stress analyses) and acceptance criteria for the normal conditions given in Tables 6.1 and 6.2 are intended to prevent bolt failures by incremental or progressive plastic deformation or by low-cycle fatigue. They are also expected to preserve the overall elastic behavior of the closure bolt so that the bolt preload and the leak-proof seal of the bolted closure can be maintained.

The analysis (the maximum stress analysis) and criteria for the accident conditions given in Table 6.3 are intended to prevent failures by excessive plastic deformation or by the rupture of the bolt. Using the yield stress as the stress limit for average tensile bolt stress implies that a small amount (.02%) of plastic deformation is permitted. Therefore, these stress criteria are not intended to preserve the full preload after an accident condition. If the full preload is needed to prevent leaks after an accident, lower stress limits must be used.

For the accident condition, the deformation-controlled secondary stresses (like the bending and residual torsional stresses) become less significant in the determination of bolt failure because the magnitude of these secondary stresses can be drastically reduced by the relatively larger bolt deformation permitted for the accident condition. Therefore, it is justifiable to ignore these secondary stresses for the accident condition. As shown in Appendix II, the ASME B&PVC, Section III does eliminate the consideration of bending and residual torsional stresses for accident conditions except in the case of bolts having tensile strength greater than 100 ksi.

6.4 Analysis Procedure

A closure bolt stress analysis that meets the requirements and criteria specified in Subsection 6.1 of this report involves the following major steps:

(1) Identification of individual loadings.
Referring to Subsections 3.1 through 3.2 and Subsection 6.1, identify each of the bolt and cask loadings to be included in the closure bolt analysis.

(2) Identification of critical combined load cases.
Place all concurrent loadings into a group and combine the extreme conditions of these loadings in all possible ways to create possible critical combined load cases for the closure bolt analysis.

(3) Identification and evaluation of load parameters.
Referring to Subsections 4.2 through 4.8, identify for each individual loading, the load parameters needed for the evaluation of closure bolt forces/moments. Examples of the load parameters would include the maximum impact acceleration (ai) and the dynamic load factor (DLF) for an impact load, the stress-free temperature, and the maximum temperature for a temperature load. For each of the load parameters, assign an appropriate value for the combined load case to be evaluated.

(4) Determination of bolt forces/moments of individual loading.
Using the applicable formulas from Tables 4.1 through 4.8, find the bolt forces/moments and the fixed edge force (Ff) and moment (Mf) for each of the loadings of the combined load case to be analyzed.

(5) Determination of bolt forces/moments of combined load case.
Following the procedure given in Table 4.9, combine the bolt forces/moments of all the loadings in the combined load case to obtain the bolt forces/moments of the load case.

(6) Evaluation of bolt stresses of combined load case.
Using the appropriate formulas from Table 5.1 and the bolt forces/moments of the combined load case, obtain the average and maximum bolt stresses required for comparison with the criteria.

(7) Comparison with acceptance criteria.
Compare the obtained stresses of the combined load case with the the stress criteria specified in Subsection 6.1 of this report. If the criteria are not met, the bolt design is not acceptable and an analysis of the remaining load cases is not needed.

6.5 Suggestions to Facilitate Analysis

The closure bolt analysis effort can be greatly facilitated if the more critical cases can be identified and analyzed first. To help the analyst, this subsection provides some insight into the relative importance of the various stress and loading conditions.

Between the maximum stress and the fatigue stress analyses, the maximum stress analysis normally controls the closure bolt design. Between the normal and accidental conditions, the accident conditions usually dominate. Among the accident conditions, the fire condition and the free drop condition are the most severe. Among the normal conditions, the preload condition and the free drop condition will prevail.

Regarding fatigue analysis, the most significant loading is probably the repeated preload. Excessive prying and bolt bending are also concerns. Significant vibration loadings occur only at resonance and must be eliminated by design modification—the analysis method described here is not adequate for high-cycle fatigue analysis. As to the cyclic pressure and temperature loads, the ASME B&PVC (Ref. 3) specifies that for a nuclear pressure vessel stress analysis, a fatigue analysis of a bolted joint is not needed unless the adjacent components need one.

Comparing the prying and bending actions, the former produces more significant bolt stresses than the latter. Significant prying can be generated by impacts especially when the impact load has a magnitude comparable to the bolt preload. The prying effect is caused by the relative rotation of the closure the lid and the cask wall. It can be effectively minimized by two basic approaches:

(1) Reduce the fixed-edge moment (Mf) generated by the applied load by relocating the load closer to the bolt circle.

(2) Reduce the relative rotation of the closure lid and the cask wall by stiffening or thickening these components (especially in the unsupported areas of the component such as the center area of the closure lid).

As shown in the study in Appendix III, one of the most effective ways to reduce prying is to increase the closure lid thickness. Increasing the bolt size and changing the preload have uncertain results. Using the plate-plate model developed in Appendix III, one can also show that a closure lid having two different thicknesses produces much less prying action than a lid of a uniform thickness. A closure lid having a smaller flange thickness (like the lid shown in Fig. III.6 of Appendix III and the lid with a bored seal shown in Fig. 2.1 of this report) is especially effective in reducing the prying tensile bolt force.

7.0 DESIRABLE ENGINEERING PRACTICES CONCERNING CLOSURE BOLTS

The use of the stress analysis and acceptance criteria presented in this report will encourage desirable practices in the design, operation, and maintenance of bolted closures for shipping casks. Some of the desirable practices are as follows:

- Use the protected closure lid and bolt to avoid large shear bolt load caused by the direct impact of the closure lid or bolt.
- Use the shear lip or the keys in the closure lid to reduce the shear load on bolts. As pointed out in the preceding subsection, a closure lid with a shear lip (i.e., a closure lid with a thicker central area over the cask cavity) also significantly reduces the prying tensile bolt force.
- Use one or more of the following methods to minimize bolt prying and bending which can cause excessive bolt stress and bolt fatigue:
  - Use a sufficiently thick or stiffened closure lid.
  - Use a closure lid diameter that is as small as possible.
  - Locate the applied load as close to the cask wall as possible.
  - Protect the closure lid from the large applied load.
  - Isolate the bolted closure joint from rotation and bending moment transmitted through the closure lid and the cask wall.
  - Use adequate preload to minimize leakage and fatigue, but avoid setting a preload value near the magnitude of the dominant or critical applied load.
- Use anti-vibration-loosening devices or other methods to maintain a steady operating preload.
- Use gaskets whose ability to maintain the seal of the bolted closure does not vary significantly with changes in closure bolt force and preload.
- Minimize conditions like the misalignments of components or large bolt hole clearances that can lead to significant bolt bending.
- Use materials, gaskets, lubricants, and practices described in Section 2.5 to minimize friction and preload variations.

Appendix II of the ASME B&PVC, Section III (Ref. 3) provides additional information concerning the design and analysis of bolted joints.

8.0 QUALITY ASSURANCE

Regulations in 10 CFR 71, Subpart H require that Type B packaging be designed, constructed, and maintained under a certified quality assurance (QA) program. Documented procedures and practices should be in place and used to implement the QA program. Every effort should be made through the procedures to eliminate the possible use of counterfeit or bogus bolts which do not meet the specified design standards for the closure bolts. In addition, the practices and procedures should address the qualification, acceptance, and preshipment testing requirements for the closure bolts.

The bolts must be designed and qualified to meet the normal and accident conditions for radioactive material containment as specified in 10 CFR 71. The qualification testing of the bolts should include the qualification of their preload requirements and tightening methods. The number of times the bolts can be tightened for shipment during their life cycle and the expected environment should also be qualified by testing.

Bolts should undergo acceptance testing prior to their first use in a containment system. The acceptance tests should include both destructive and non-destructive testing, independent of the supplier to preclude the use of counterfeit and bogus bolts. The non-destructive tests should include inspection for cracks, burrs, defects, surface finish, proper dimensions, and material hardness. Destructive tests should include strength, brittle fracture, and chemical composition testing. The sample sizes for non-destructive and destructive tests should consider lot size, maximum acceptable percentage of defects, and confidence level (Refs. 10–12).

Prior to each shipment, each bolt and its counterpart should be visually checked for abnormal wear or damage to insure that they meet the specified minimum design requirement throughout their life cycle. Bolts not meeting the minimum requirements should be removed and replaced. Replacement bolts should meet all the original design qualification and acceptance requirements. All testing and assembly of the bolting closure should be performed by a qualified mechanic with the proper tools under the QA program.

9.0 CONCLUSIONS

This report has shown that the structural behavior of bolted joints can vary significantly with the designs and applications of the bolted joints and components. Existing structural analysis methods and industrial codes for bolted joints must not be applied indiscriminately to shipping casks without taking into consideration the differences in design and application.

Large flat closure lids, extreme fire temperatures, severe impact loads, and strict leak-proof requirements are some of the unique conditions in the design and application of shipping casks. The large flat closure lids can produce appreciable prying and bending of the closure bolts. The extreme fire temperatures can cause an excessive tensile bolt force in addition to the bolt preload. The severe impact load can increase the risk of fractured bolts. The strict sealing requirements can limit the allowable permanent deformations of the closure bolts.

The stress analysis procedures and the formulas and criteria developed in this report are intended to address these special concerns for the closure bolts of shipping casks. Appendix VII is a bibliography of information on bolted joints which can be used to supplement this report.

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APPENDICES

APPENDIX I

Structural Behavior of Bolted Closure Joints

1.0 Introduction

The structural behavior of a bolted closure joint is the result of the mechanical interaction of the joint components (i.e., the closure lid, the closure bolt, and the cask wall (see Fig. I.1.)). The gasket has a secondary role; its main function is to prevent leakage. In general, the interaction of the joint components is complicated and involves geometric arrangement and tolerance, and the local and global deformations of the joint components. To provide the necessary understanding of the analysis methods and criteria described in this report, this appendix highlights some of the pertinent information found in the literature on this subject. Specifically, this appendix attempts to show; (1) how the local and global deformations of the joint components affect the structural behaviors of the bolted joint and the forces in the bolts, (2) what are the dominant bolt forces, and (3) what parameters determine the bolt forces.

2.0 Effect of Closure Lid Deformation on Bolted Joint Behavior

To demonstrate the possible effect of joint-component deformation on the structural behavior of a bolted joint, Fig. I.2 depicts the different behaviors of three bolted closure designs under the action of an applied axial tensile load. The three designs, A, B, and C are identical except in the thickness of the closure lid. Design A has the largest closure-lid thickness and Design B has the smallest. Since the lid of Design A is very thick and stiff it can probably lift the bolt heads up causing little moment resistance at the bolted joint. Thus, the bolted joint of this design would appear to a be a roller or a hinged joint. Design B, on the other hand, has a very thin closure lid which is significantly more flexible than the closure bolts. With sufficient tensile preload, the stiff closure bolts will be able to clamp the lid down and allow little rotation and separation of the lid to appear at the bolted joint. Thus, the bolted joint of this design would appear to be a rigid joint. Design C has a closure lid stiffness that is comparable to the closure bolts. The closure lid of this design can lift the bolt heads up, but in the process the lid itself also shows appreciable deformation and rotation at the bolted joint. The lifting of the bolt heads is actually accomplished with the help of a prying action. With the partial opening of the joint and the rotation of the lid relative to the cask wall, the bolted joint of this design would appear to be a semi-rigid joint.

The above example clearly shows that the joint behavior depends on the deformation of the joint components. Therefore, the determination of the bolt forces and the forces and moment transmitted through the bolted joint must include a due consideration of the component deformation. As an example, Appendix III analyzes the prying action of Design C in further detail and provides approximate formulas for estimating the tensile bolt force caused by prying.

3.0 Distribution of Load Among Bolts

The deformation of joint components also has an effect on the distribution of an applied load to all the bolts in the bolted joint. Naturally, bolts located in an area with a greater component deformation tend to receive a greater share of the applied load. This phenomenon is depicted in Fig. I.3. This figure shows how the closure-lid deformation can affect the distribution of applied tensile and shear loads to more than one row of closure bolts. Experience has shown that the row of bolts closest to the load would receive a greater share of the load because of the non-uniform deformation of the closure lid.

The distribution of the load to bolts can also be significantly affected by factors other than joint-component deformation. As shown later in this appendix, the bolt forces are affected by the residual bolt preload and the design tolerances of the joint components. The preload and tolerances are either difficult or impossible to control. In practice, for various reasons the actual preload can vary as much as ±30% of the design preload. The tolerances are attributed to the imprecision of the fabrication process. Thus, it is futile to pursue a detailed analysis of the precise distribution of the applied load among the bolts including all the possible effects of component deformation, preload, and design tolerances. For the sake of simplicity it is customary to ignore all these effects in the design analysis of bolted joints and obtain essentially an average magnitude of the bolt force. To calculate the bolt force from the applied load, the closure lid is treated as a rigid plate and all the bolts are considered equally effective in resisting the load. Thus, the distribution of a load to the bolts is determined only by the direction and location of the load relative to the bolts. Using this approach, the bolts in Figs. I.2 and I.3 will be shown to share the load equally. The average bolt force obtained using this simplified approach should be adequate for the analysis of phenomena involving large bolt deformation such as the ductile failure of bolts. However, for phenomena involving small bolt deformation such as the leakage of bolted joints, the brittle fracture of bolts, and the high-cycle fatigue of bolts, the results of the simplified approach should only be used with great caution and conservatism.

4.0 Bolt Forces

A closure bolt is basically a one-dimensional structural member. It is most resistant to axial tensile and transverse shearing deformations. Bolted joints are designed to take advantage of these desirable bolt characteristics. A bolted joint, regardless of the load it supports, derives its strength primarily from the shear and tension load-carrying capacities of the individual bolts. For this reason, past studies of the behavior of bolted joints have been focused on the analysis and measurement of the axial-tension and transverse-shear performance of the bolts in a bolted joint. As indicated in Fig. I.1, bending and torsional moments (or stresses) can also exist in the bolts and help directly or indirectly to support the applied loads. However, they have been shown to play a secondary role in the determination of the strength and performance of the bolted joint. Accordingly, they have received relatively less attention in the literature on this subject. Each of the bolt forces will be discussed in the following sections.

4.1 Tensile Bolt Force Due to Applied Tensile Load

A tensile bolt force is basically generated by an applied tensile load. However, it can also be generated by a prying action and its magnitude is significantly affected by the preload in the bolt. To maintain a tight joint and for other reasons, a tensile preload is always applied to the bolts of a bolted joint. Thus, it is essential to understand the role of the preload and include it in the calculation of the bolt forces.

Figure I.4 presents several typical tensile bolt force-load relations that are obtained with various bolt preloads. Three curves are shown and each is for a given tensile preload, P_1, P_2, or P_3. The curves show how the tensile force (F) of a bolt increases with a concentric tensile load applied at the bolted closure (L). At zero applied load, the bolt force should be equal to the preload. As the tensile applied load increases, the tensile bolt force also increases but at a rate much smaller than the applied load. The rate of increase of the bolt force remains low until the outer edge of the bolted joint starts to separate. After this incipient separation, the rate of increase of the bolt force starts to increase and eventually approaches the rate of increase of the applied load when the joint is completely separated. The cause for this change of bolt force is illustrated in the bolted-joint drawings located above the bolt force-load curves. The drawing corresponding to the zero applied load shows that a distributed compressive stress exists at the interface between the closure lid and the cask wall. This stress is caused by the bolt preload and is required to maintain the equilibrium with the bolt preload. The resultant force of the compressive joint stress must, therefore, be equal in magnitude to the bolt preload P when there is no applied load. As soon as a tensile load is applied to the closure lid, the compressive stress is reduced because the applied load lifts the lid up residual bolt preload and the design tolerances of the joint components. The preload and tolerances are either difficult or impossible to control. In practice, for various reasons the actual preload can vary as much as ±30% of the design preload. The tolerances are attributed to the imprecision of the fabrication process. Thus, it is futile to pursue a detailed analysis of the precise distribution of the applied load among the bolts including all the possible effects of component deformation, preload, and design tolerances. For the sake of simplicity it is customary to ignore all these effects in the design analysis of bolted joints and obtain essentially an average magnitude of the bolt force. To calculate the bolt force from the applied load, the closure lid is treated as a rigid plate and all the bolts are considered equally effective in resisting the load. Thus, the distribution of a load to the bolts is determined only by the direction and location of the load relative to the bolts. Using this approach, the bolts in Figs. I.2 and I.3 will be shown to share the load equally. The average bolt force obtained using this simplified approach should be adequate for the analysis of phenomena involving large bolt deformation such as the ductile failure of bolts. However, for phenomena involving small bolt deformation such as the leakage of bolted joints, the brittle fracture of bolts, and the high-cycle fatigue of bolts, the results of the simplified approach should only be used with great caution and conservatism.

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The division of applied load between the maintenance of the preload and the increase of the bolt force is determined by the stress-deformation characteristics of the bolt and the joint. Simple formulas can be derived for this purpose using a model of two parallel springs of different lengths. As shown in Fig. I.5, the shorter of the two springs represents the bolt and the other simulates the joint. The two springs are forced to have the same length by welding them together at the two ends. After the welding, the bolt spring will develop a tensile preload and the joint spring will have a compressive force. Similar to the bolt preload and the compressive joint force in a bolted joint, the two spring forces will be equal in magnitude. When a tensile load increment (dL) is applied to this two-spring system at the ends, the two springs will be stretched the same amount (dx), and the stretch will, in turn, induce an increase of the tensile bolt-spring force (dF) and a reduction of the joint-spring compressive force (dC) just as in a bolted joint. All of these changes of forces and deformation are related as follows:

$$dF + dC = dL \qquad (I.1)$$

$$dF = k_b \, dx \qquad (I.2)$$

$$dC = k_j \, dx \qquad (I.3)$$

where kb and kj are the bolt and joint spring constants or stiffnesses respectively. The change of spring length (dx) can be eliminated in these three equations to produce the following relations between the changes of applied load and spring forces:

$$dF = \frac{k_b}{k_b + k_j} dL \qquad (I.4)$$

$$dC = \frac{k_j}{k_b + k_j} dL \qquad (I.5)$$

If the bolt and joint stiffnesses are constants, Equation I.4 represents a linear relation between the applied load and the bolt force. Test results of actual bolted joints do show that an approximately linear relation between the bolt load and force holds until the joint starts to separate, and thus support the modeling of the joint and bolt as linear springs. Figure I.6 compares a typical bolt force-load relation with the linear relation of the simple two-spring model. After the joint starts to open and before it is fully separated, the contact area between the closure lid and the cask wall decreases continuously and causes the actual bolt load-force curve to show a rapid and nonlinear increase of the bolt force with increasing applied load. This nonlinear segment can be clearly identified in the typical bolt load-force curves shown in Fig. I.4.

In most bolted-joint designs, the joint is significantly stiffer than the bolt; the joint stiffness is usually more than five (5) times the bolt stiffness. Based on this information, Equations I.4 and I.5 show that for most bolted joints, the joint compression should disappear and the joint should be fully open when the applied load and the bolt force reach 120% of the bolt preload. This result (which is shown in Fig. I.6) implies that for most bolted joints, an applied load whose magnitude is less than the bolt preload can never cause the bolt force to exceed the bolt preload by an amount more than 20% of the preload. This small difference between the bolt preload and the bolt force is the basis for a simple rule that is commonly used for bolt force calculation; i.e., the tensile bolt force is set equal to either the preload or the applied load depending on whether the applied load is below or above the preload, respectively. Analysis methods described in this report use this simple rule for the determination of the tensile bolt force. Figure I.6 also compares this simplified bolt force-load relation to the relation obtained using bolt and joint stiffnesses.

4.2 Tensile Bolt Force Due to Prying

If the axial applied load is not aligned with the bolt axis, the load and the bolt force will form a moment, which in turn will cause the closure lid to rotate relative to the cask wall. This rotation can sometimes produce an additional tensile bolt force through a prying action. Figure I.7 shows how the prying action can be generated. It also shows that when a prying action is present, an additional tensile bolt force is caused by the reaction of the cask wall (R). The most critical case shown therein is Case b.1 where the additional bolt force is added to the bolt force caused by the applied load. Figure I.8 shows a typical set of bolt force-load curves for bolted joints with and without the prying action. The information present in this figure and Fig. I.2 suggests that the prying force depends on many factors including the applied load, the bolt preload, and the deformations of the bolt and the closure lid. This dependence of the prying action on the closure lid deformation will be discussed further in this appendix. Approximate formulas are developed in Appendix III for the evaluation of the bolt force due to prying.

As shown in Fig. I.9, an applied shear load can also create a prying action and thus produce a tensile bolt force. However, unless the closure lid is thick and the shear load is large, the additional tensile bolt stress caused by the shear load will be insignificant.

4.3 Tensile Bolt Force Due to Cyclic Axial Load

The fluctuation of the tensile bolt force due to an applied cyclic tensile load can be analyzed using the bolt force-load curves presented in Fig. I.8. Figure I.10 shows the results for two cases where the maximum applied load is below and above the bolt preload, respectively. For each case, the effect of prying is also demonstrated. It is shown therein, when the applied load varies from a zero to a maximum value, the bolt force changes from the initial preload to a maximum value corresponding to the maximum applied load. The maximum bolt force is determined by the respective bolt-force load curve.

Comparing the cases with and without prying in Fig. I.10 for the same bolt preload and applied load, the case with prying always has a higher maximum bolt force and thus shows a greater amplitude of fluctuation of bolt force than the case without prying. This result is demonstrated by the time histories shown on the right-hand side of Fig. I.10. Since a greater fluctuation of the bolt force can induce greater fatigue damage in the bolt, the presence of a prying action is clearly detrimental to the fatigue resistance of the bolt. Therefore, to reduce the risk of bolt fatigue, a bolted-closure design should minimize the prying action. The detrimental effect of prying action on the fatigue of bolted joints has been demonstrated experimentally by several investigators.

Contrary to the prying action, the bolt preload can drastically reduce the fluctuation of the bolt force and thus help reduce fatigue damage. This effect is also demonstrated in Fig. I.10. Assuming no prying action, compare the two cases with the preload below and above the applied load, respectively. The case with the preload above the applied load clearly shows a smaller fluctuation of the bolt force in response to the same fluctuation of the applied load. Thus, the use of a higher preload reduces the fluctuation of the bolt force and the risk of resulting fatigue damage. As pointed out in Section 4.1 of this appendix, for most bolted-joint designs, the bolt force would not increase more than 20% from the preload if the applied load is kept below the preload. Accordingly, using a preload whose magnitude is greater than the maximum applied load, the amplitude of the bolt-force fluctuation can be kept below 20% of the applied load. The application of a preload is a common method for reducing vibration loosening and fatigue damage to bolted joints.

4.4 Shear Bolt Force Due to Applied Shear Load

A bolt preload also increases the resistance of a bolted joint to applied shear loads. As shown in Fig. I.11, the preload produces a compressive stress in the joint interface between the closure lid and the cask wall. The compressive stress in turn introduces a frictional force resisting the sliding of the lid relative to the cask wall. A shear load applied to the bolted joint must first overcome this friction before it can be fully exerted onto the closure bolts. Thus for design analysis, the simplified relation shown in Fig. I.11 between the applied shear load and the shear bolt force may be used. Similar to the relation for the tensile bolt force, Fig. I.11 shows that there is no shear force in the bolt until the applied shear load exceeds the joint friction force which is generated by the bolt preload. Once the joint friction is exceeded, the shear bolt force will have a magnitude equal to the difference of the applied shear load and the joint friction force. It should be emphasized that although the joint friction is caused by the bolt preload, the joint friction force is determined by the joint compression *not* the tensile bolt force. If a tensile load is applied to the bolted joint at the same time as the shear load, the joint compression and friction will be reduced even though the tensile bolt force may be unchanged. Moreover, since the joint stiffness is normally five (5) times the bolt stiffness as discussed earlier in this appendix, it can be shown using Equation I.5 that a large portion (nearly 80%) of the applied tensile load is used to reduce the joint compression and friction. Thus an applied tensile load is quite efficient in effecting a reduction of the joint compression and friction. This reduction of the friction force will cause the shear bolt force to appear at lower applied shear load and to have higher magnitude for the same applied shear load. Accordingly, in cases where the tensile applied load is significant compared to the bolt preload, the joint friction should be conservatively ignored and the shear bolt force simply set equal to the applied shear load.

4.5 Bending Bolt Moment

The causes for a bending moment in the closure bolt can be divided into two generic classes—geometric misalignments and applied loads. Geometric misalignments are the result of design tolerances or fabrication imprecisions. Figure I.12 shows some of the conditions that can cause bolt bending. Regardless of the cause, the bending force plays a secondary role in the determination of the structural behavior and integrity of the bolted joint. In the case of bending caused by misalignment, the bending is induced and controlled by the displacement. Similar to the thermal stress, this bolt bending stress would disappear as soon as the bolt or the constraint itself yields. In the case of bending caused by an applied force (although the bending is not induced and controlled by the displacement) the role of the bolt bending moment is limited by the bolted joint design. A bolted joint is designed to transmit a moment through the joint by bolt prying not by bolt bending. Thus, the joint behavior and integrity under an applied load is determined mainly by the tensile bolt force generated by the prying action. Accordingly, the bolt bending moment would play a significant role only when the joint deformation is small as in brittle fracture and high-cycle fatigue. In these cases it may be more effective to minimize than to evaluate the bolt bending. A precise evaluation of the bolt bending is difficult because the interaction between the bolt and other joint components is complicated. Other methods of reducing the magnitude or significance of bolt bending may turn out to be easier and more dependable than analysis. Bickford (1981) has shown that some bolt head designs can be used to minimize bolt bending.

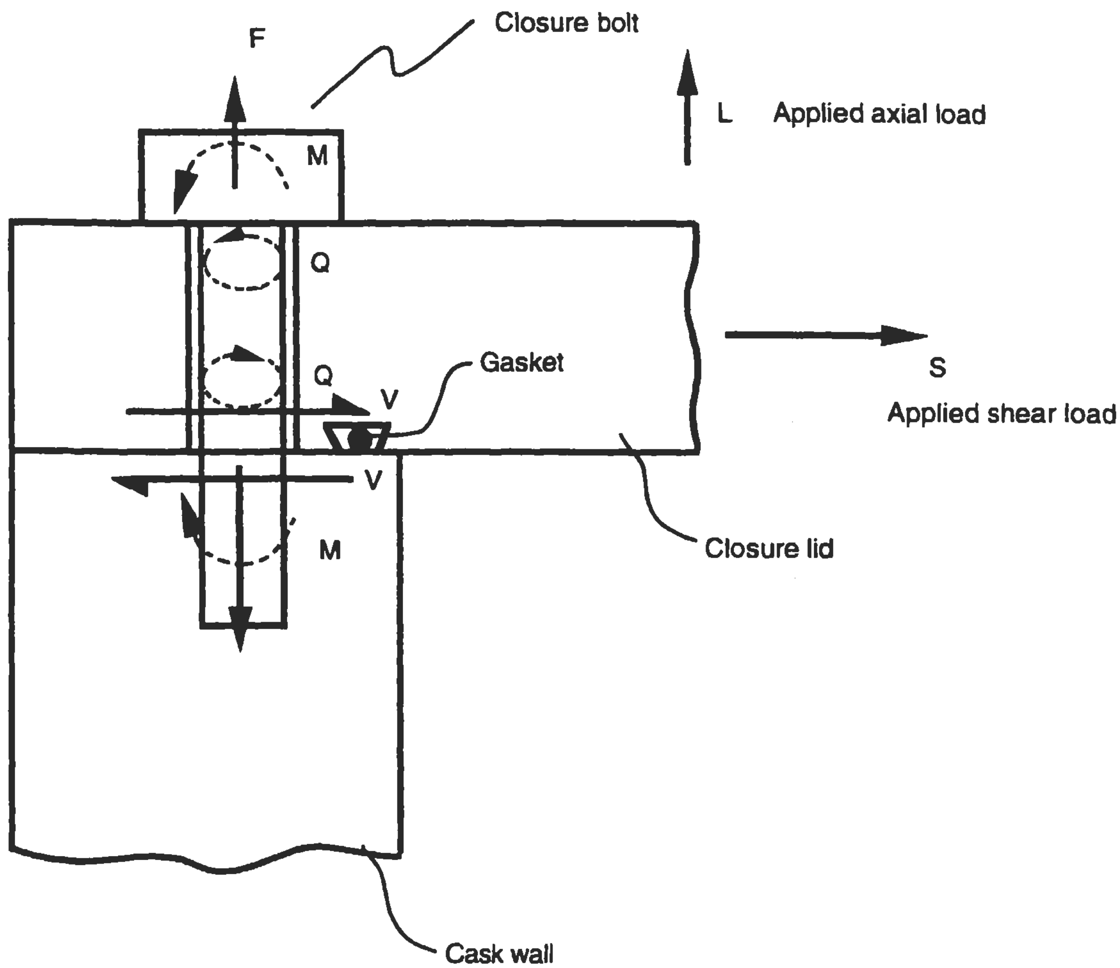
4.6 Torsional Bolt Moment

Bolted joints are not designed to produce torsion of the bolts. The bolts experience torsion only when they are preloaded with a torque wrench. A torque must be applied to the bolt head to overcome the friction between the bolt and the other joint components in order to advance the bolt into the joint and achieve the desired preload. It can be shown that for common bolt geometry about 50% of the torque is used to overcome the friction between the bottom of the bolt head and the top of the closure lid—the remainder is for the friction between the bolt threads and the cask wall. Only a small percentage (mainly 10%) of the applied torque is used to generate the preload. Although there is a definite empirical relation between the applied torque and the attained preload, experience has shown that applying preload using a torque wrench is an unreliable operation which may have an error rate as high as ±30 % if it is carried out in the field. This is mainly caused by uncertainties with the coefficient of friction which can be greatly affected by the condition of the contacting surfaces and the lubricant.

Applying preload using a torque wrench will generate a residual torsional stress in the bolt. There are greatly different beliefs concerning the possible relaxation of this stress after the preload operation. Some people believe that the residual stress disappears as soon as the preload operation is complete, while other believe that the residual stress never diminishes unless a breakaway torque is applied after the preload. Actually, the situation is more involved than this and depends on many factors. Bickford (1981) states that the amount and rate of relaxation vary substantially from bolt to bolt and from application to application. He also found that the relaxation of the residual torsional stress does not always lead to the relaxation of the residual tensile stress or prestress in the bolt. He has observed in some bolted joints that the tensile stress can actually increase 1-to-2% while the torsional stress is relaxed 50% after a preload by a torque wrench. The residual torsional stress also appears to have little influence on the ultimate tensile strength of the bolt. Tension tests of bolted joints show that the ultimate tensile strength of a bolt is not appreciably affected by the method of preload. Specimens preloaded by torque wrench show practically the same ultimate tensile strength as those preloaded by direct tension. Of course, this phenomenon can be the result of a relaxation of the residual torsional stress initiated by the large plastic extension occurring prior to the bolt failure. In fact, this deformation-initiated stress relaxation has been found to be the reason for the lack of influence of the residual bolt preload on the ultimate shear strength of a bolt in a bolted joint. Measurements of the bolt tension in a bolted joint under a shear load have shown that at the ultimate shear load, there is little preload left in the bolts. As Kulak (et al. 1987) explained, "The shearing deformations that have taken place in the bolt prior to the failure have the effect of releasing the rather small amount of axial deformation that was used to induce the bolt preload during installation." These observations concerning the residual torsional and tensile bolt stresses indicate that these stresses are deformation-controlled, secondary stresses similar to the thermal stress. Their influence on bolt failures with large accompanying deformations such as ductile failures is minimal. However, the same statement may not hold for failures with small deformations such as brittle fracture and fatigue. For the analysis of these failures, the residual bolt stresses should be regarded to be as significant as the primary stress.

References (Appendix I)

References cited in this appendix are listed in Appendix VII of this report.



Primary bolt forces/moments

F: Tensile force

V: Shear force

Secondary bolt forces/moments

M: Bending moment

Q: Torsional moment

Figure I.1 Components of a shipping-cask bolted closure and forces which may exist in a closure bolt.

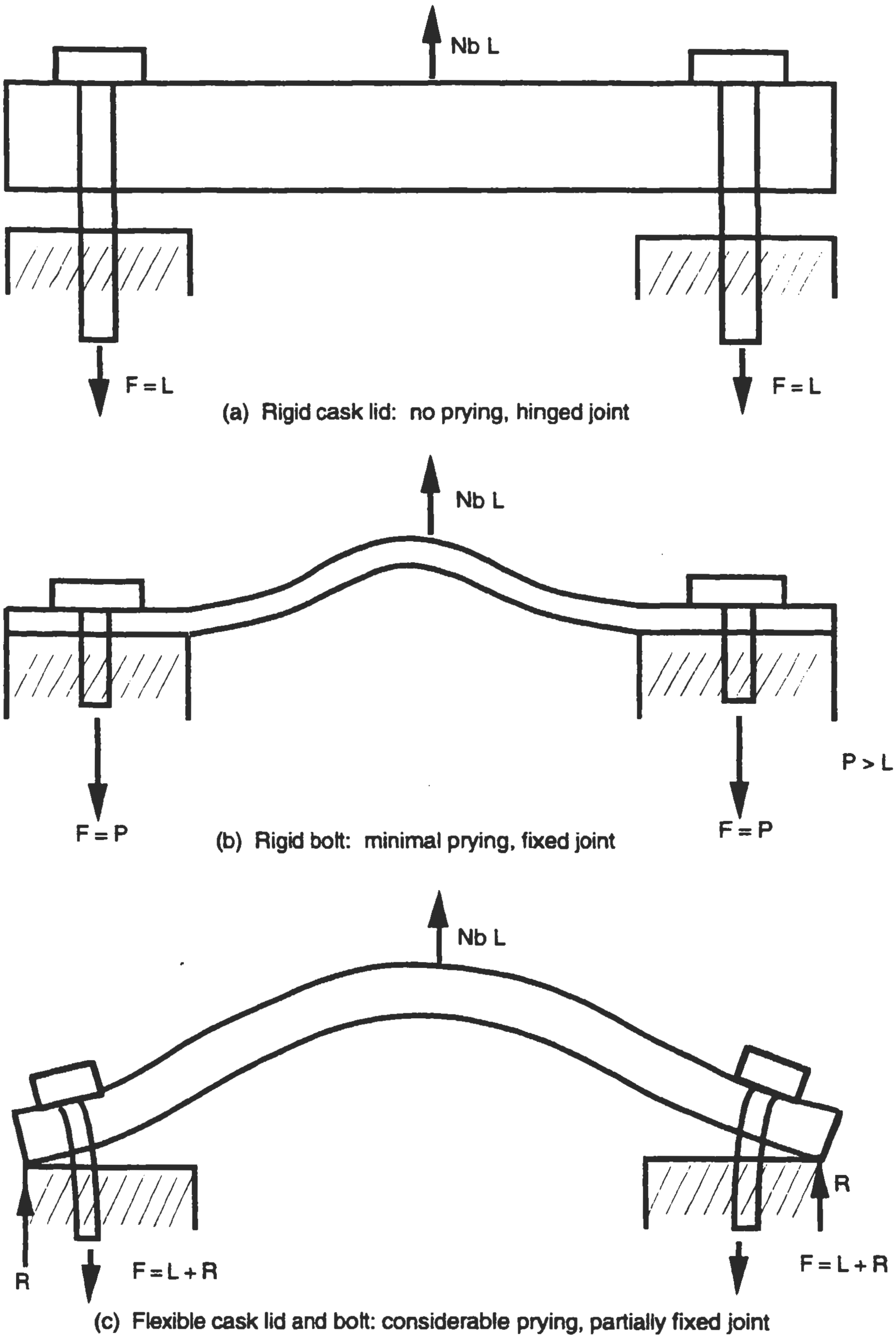


Figure I.2 The dependence of prying and joint behavior on the relative flexibility of bolted joint components.

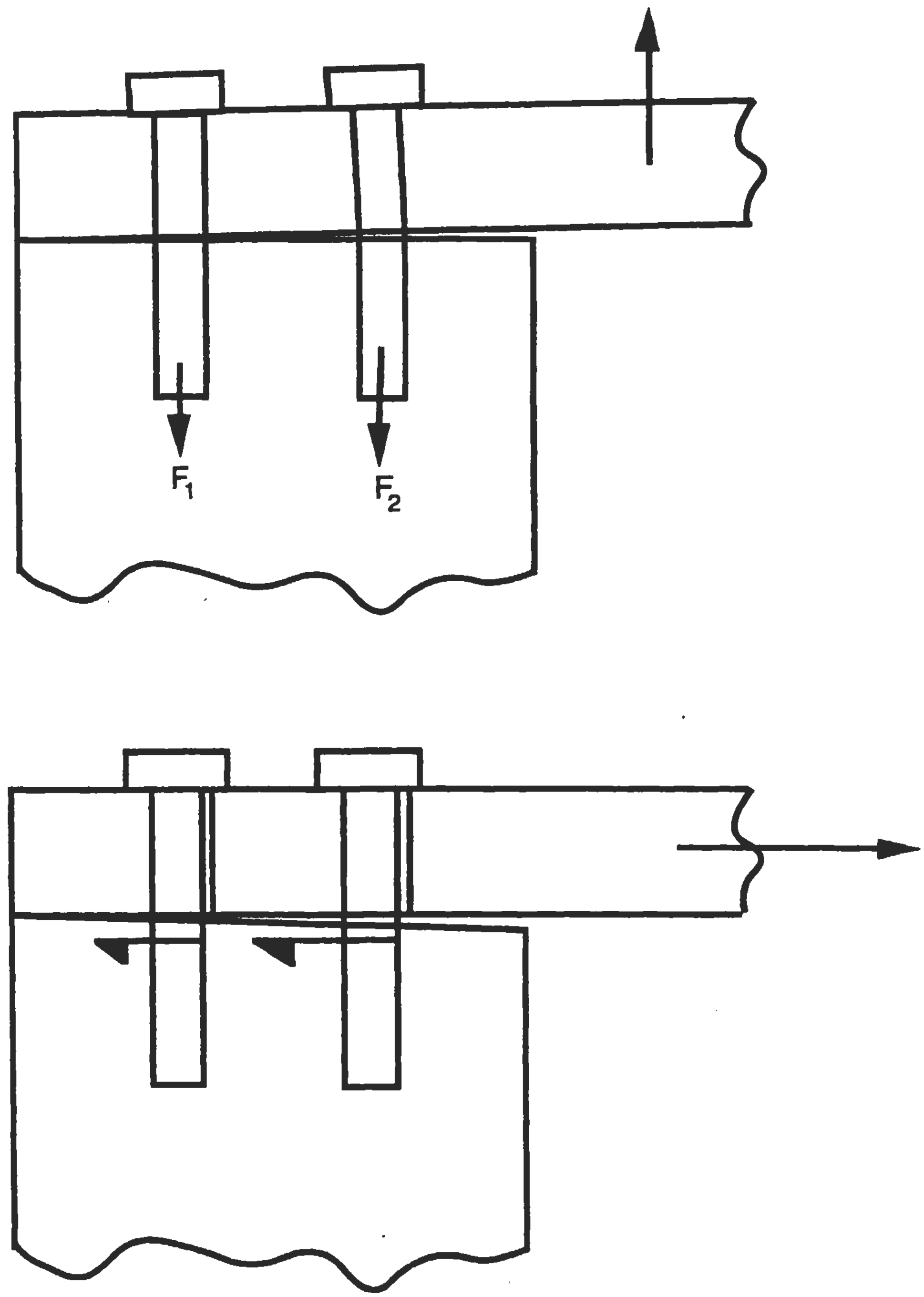


Figure I.3 Bolted closure with more than one row of bolts. (The length of the bolt force arrows indicates the probable distribution of bolt forces.)

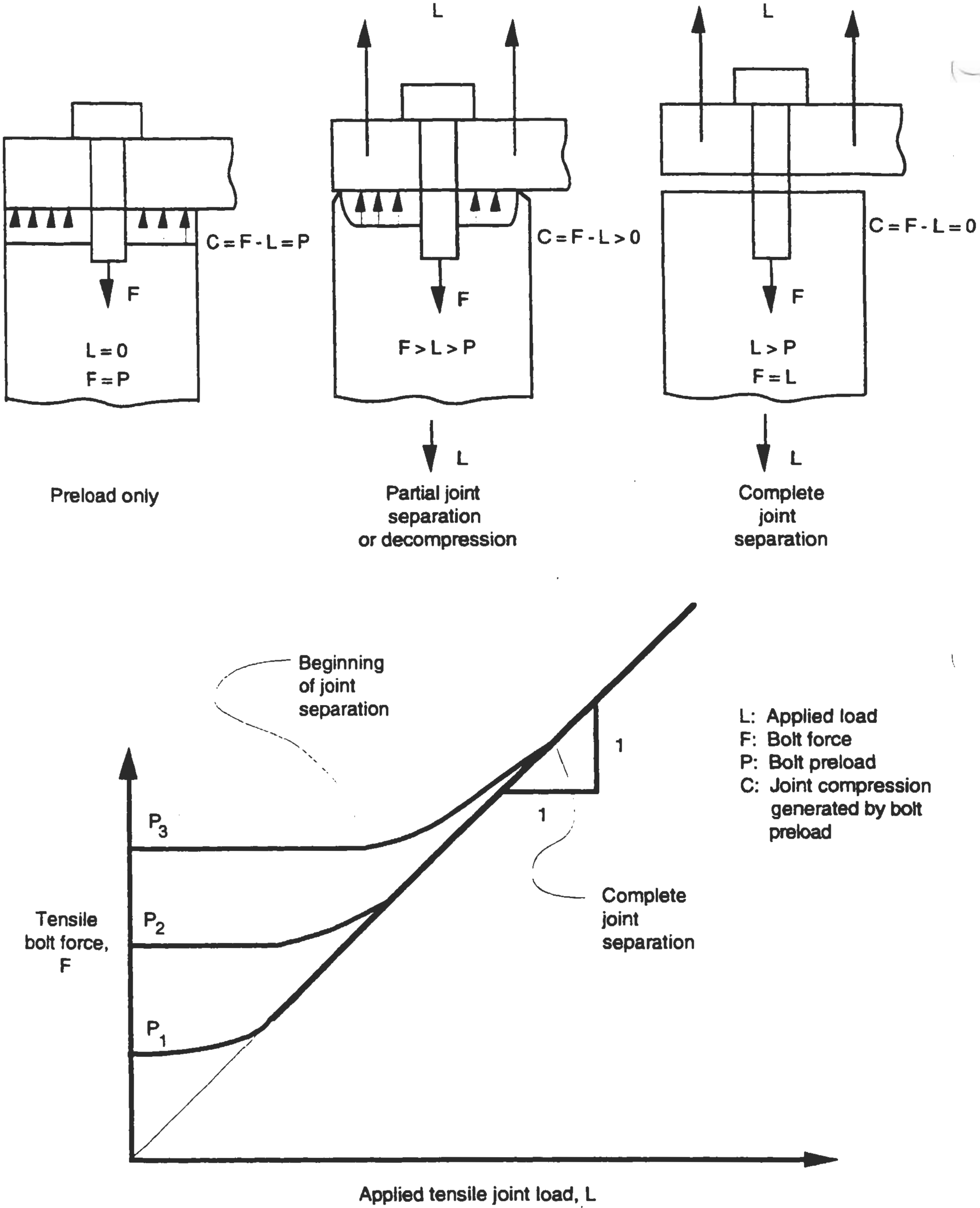
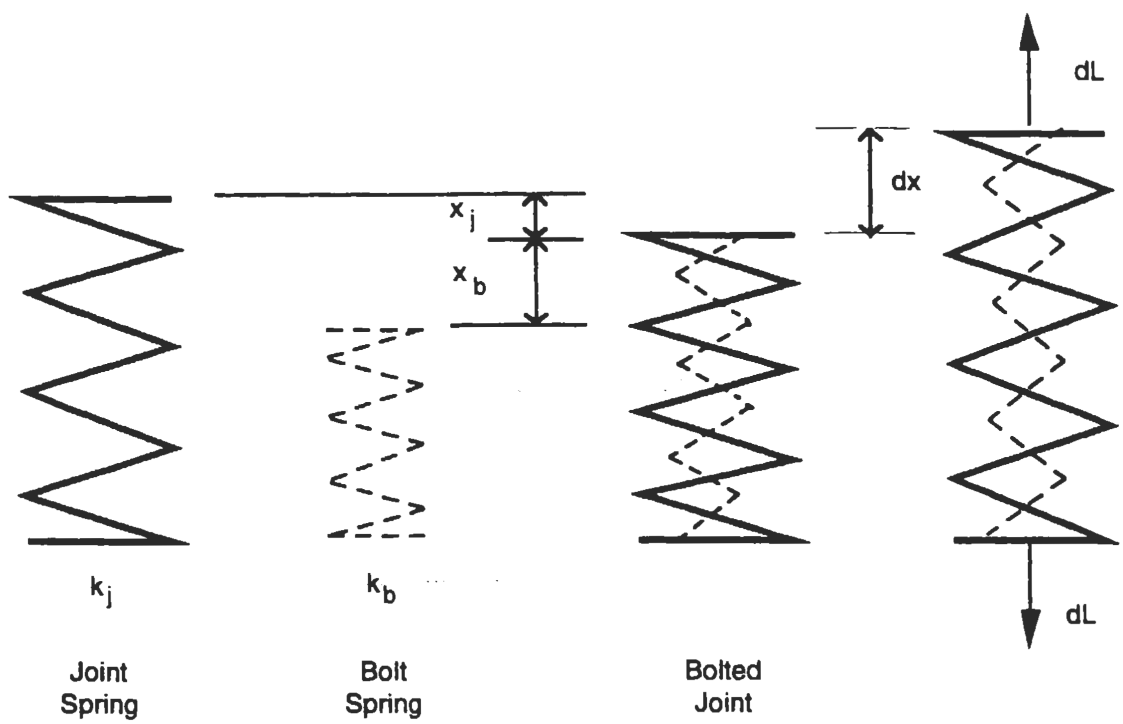


Figure I.4 Dependence of tensile bolt force on bolt preload and applied joint load.



Initial joint - spring compressive force

$$C = k_j \; x_j$$

Initial bolt-spring tensile force

$$F = k_b \; x_b$$

Figure I.5 A two-spring model of a bolted joint for analysis of tensile bolt force.

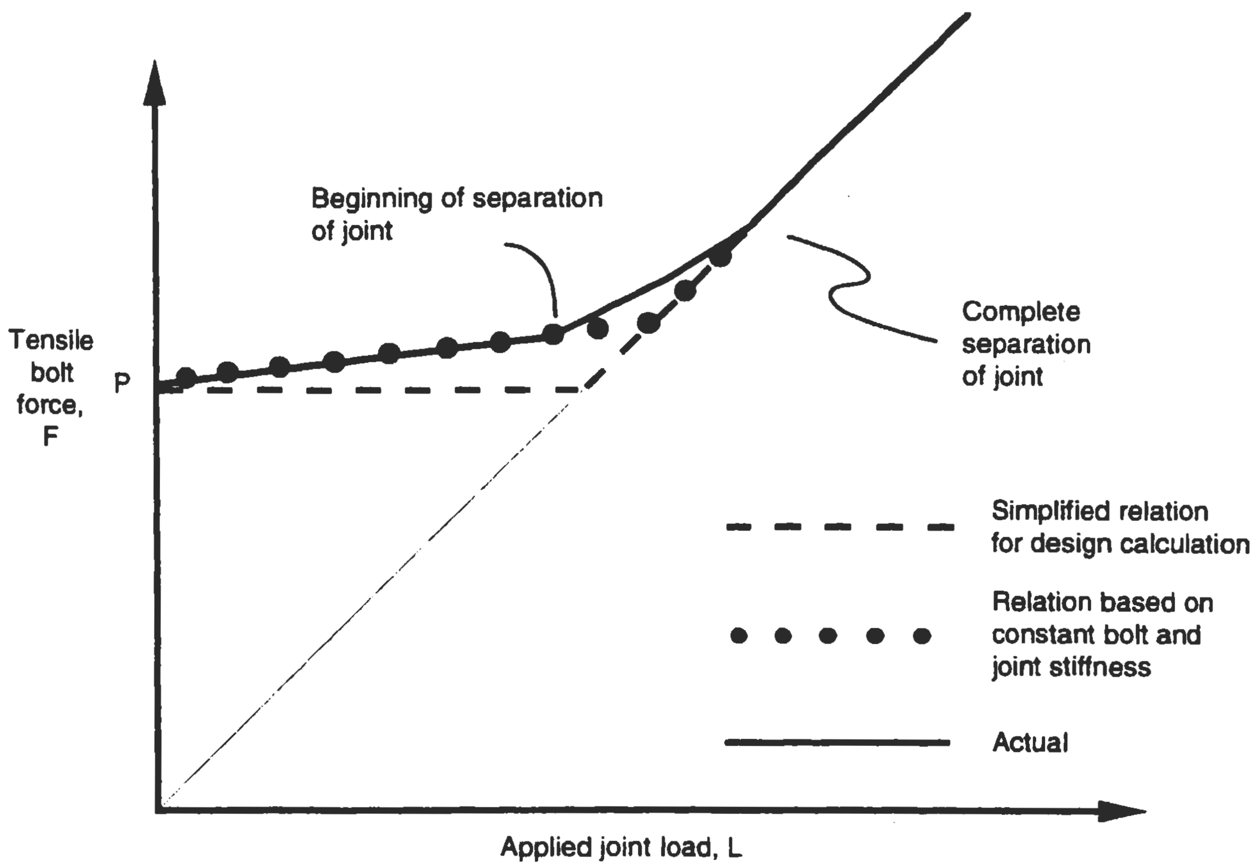


Figure I.6 Comparison of tensile bolt force-load relations.

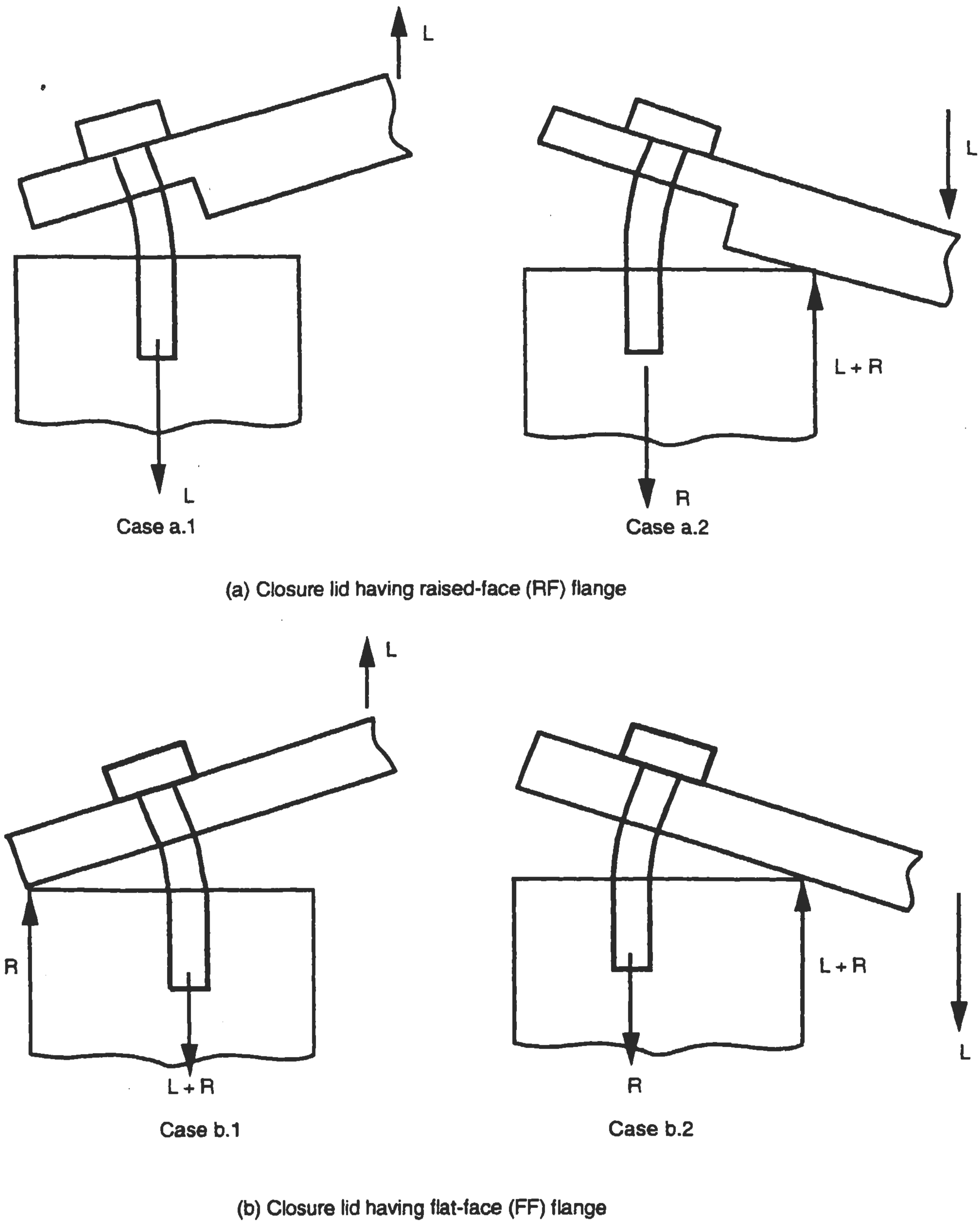


Figure I.7 Prying action caused by applied axial loads.

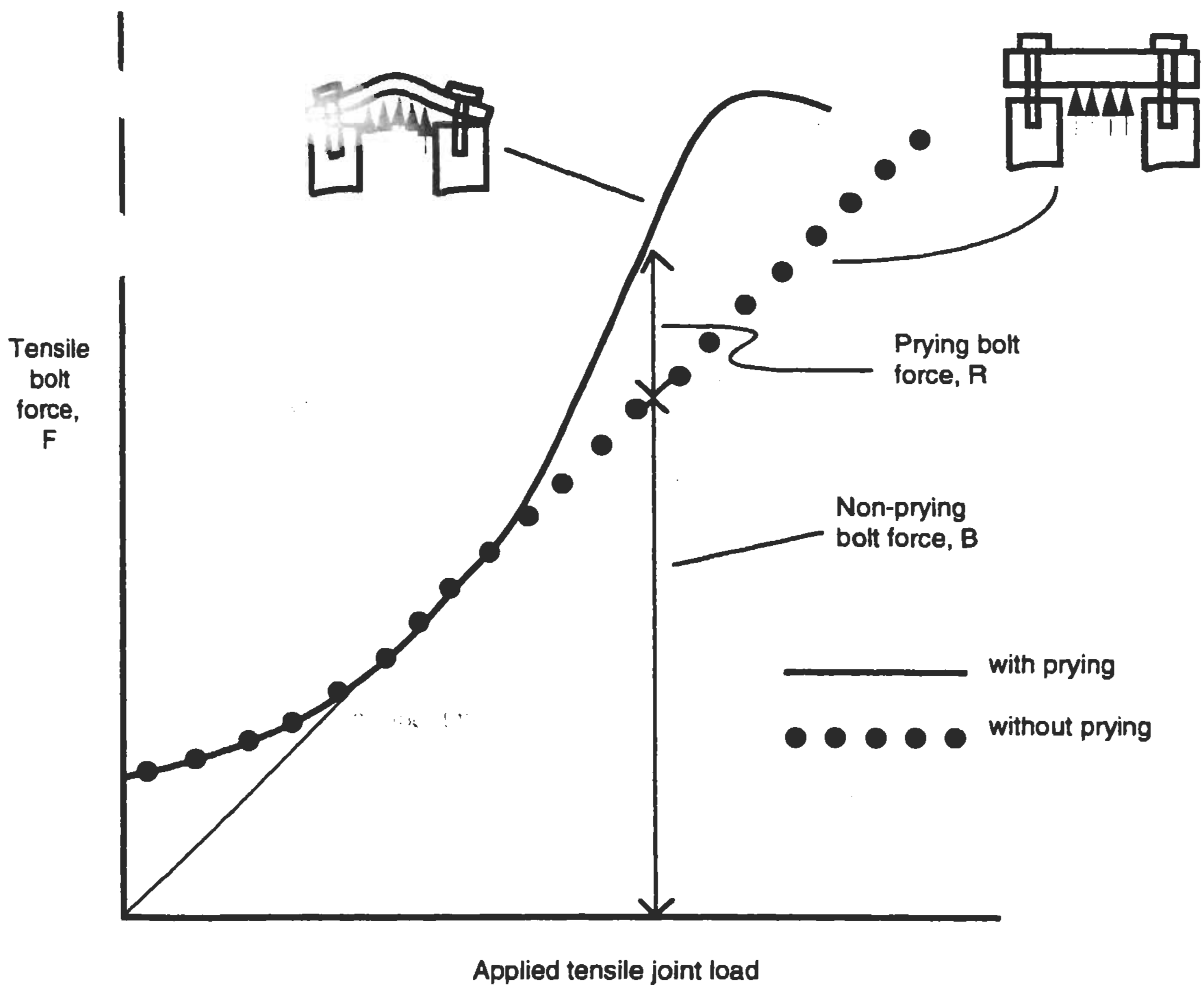


Figure I.8 Effect of prying on tensile bolt force-load relationship.

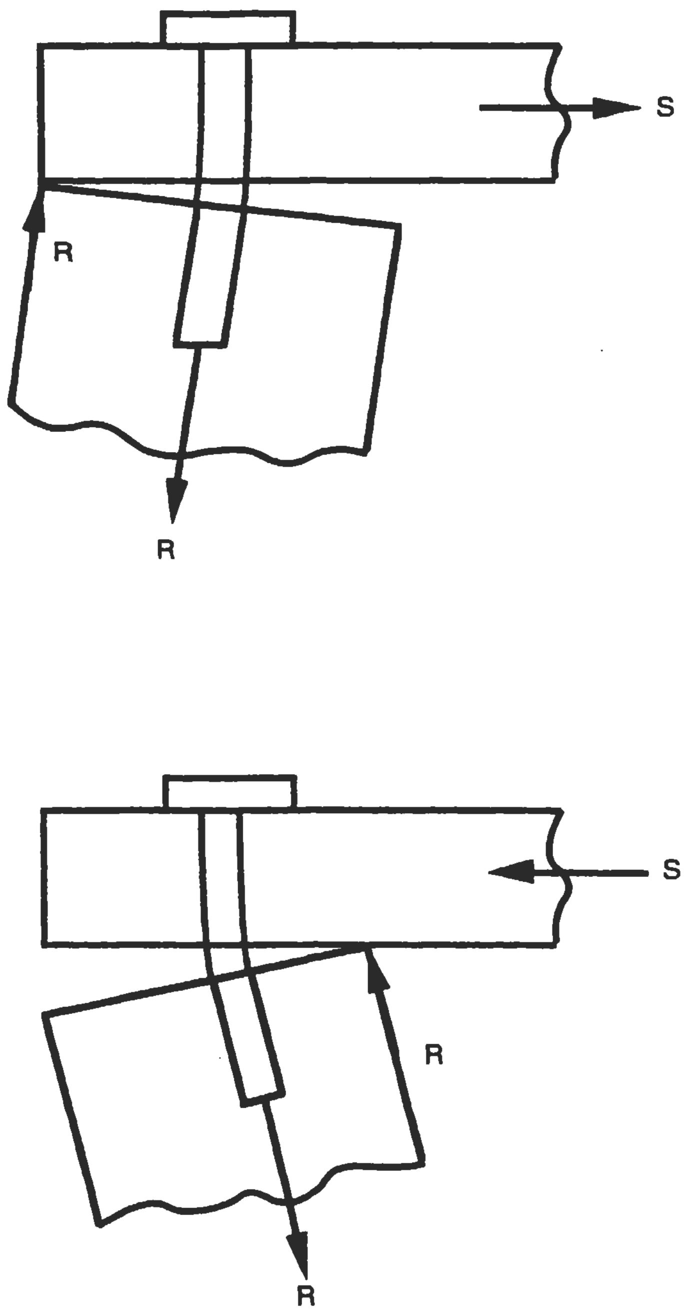
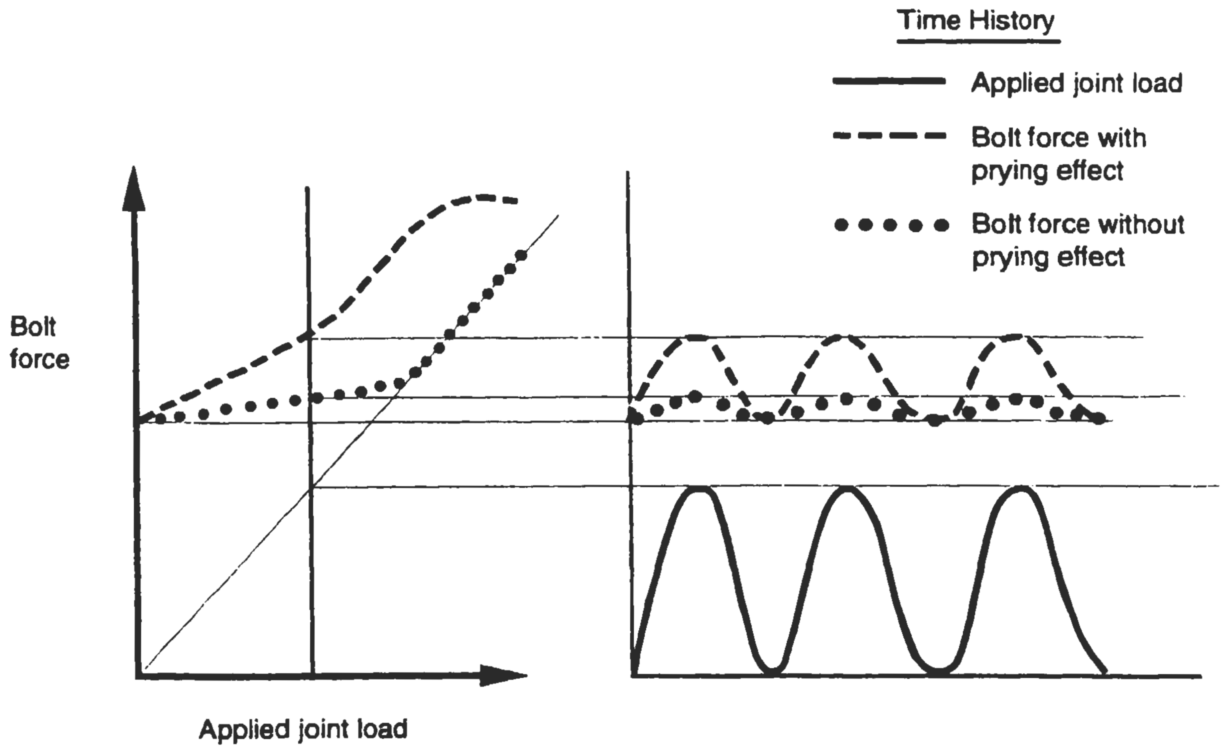
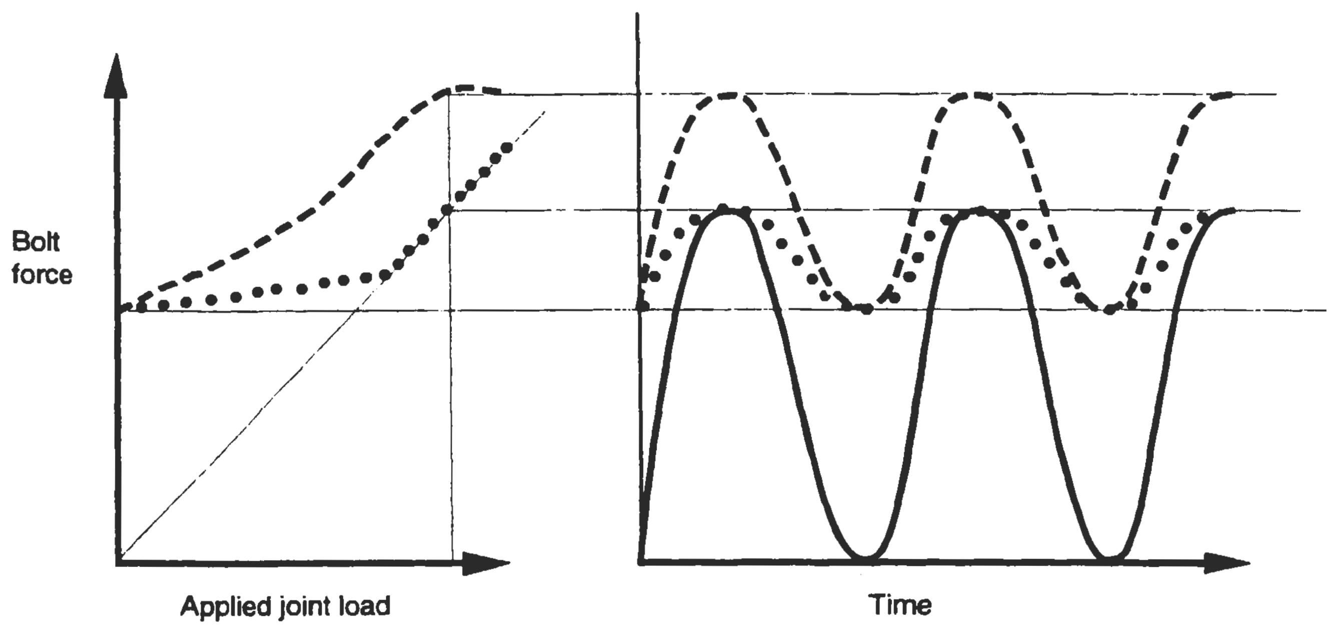


Figure I.9 Prying action caused by applied shear loads.



(a) Applied load less than bolt preload



(b) Applied load greater than bolt preload

Figure I.10 Tensile bolt forces generated by a fluctuating applied tensile load.

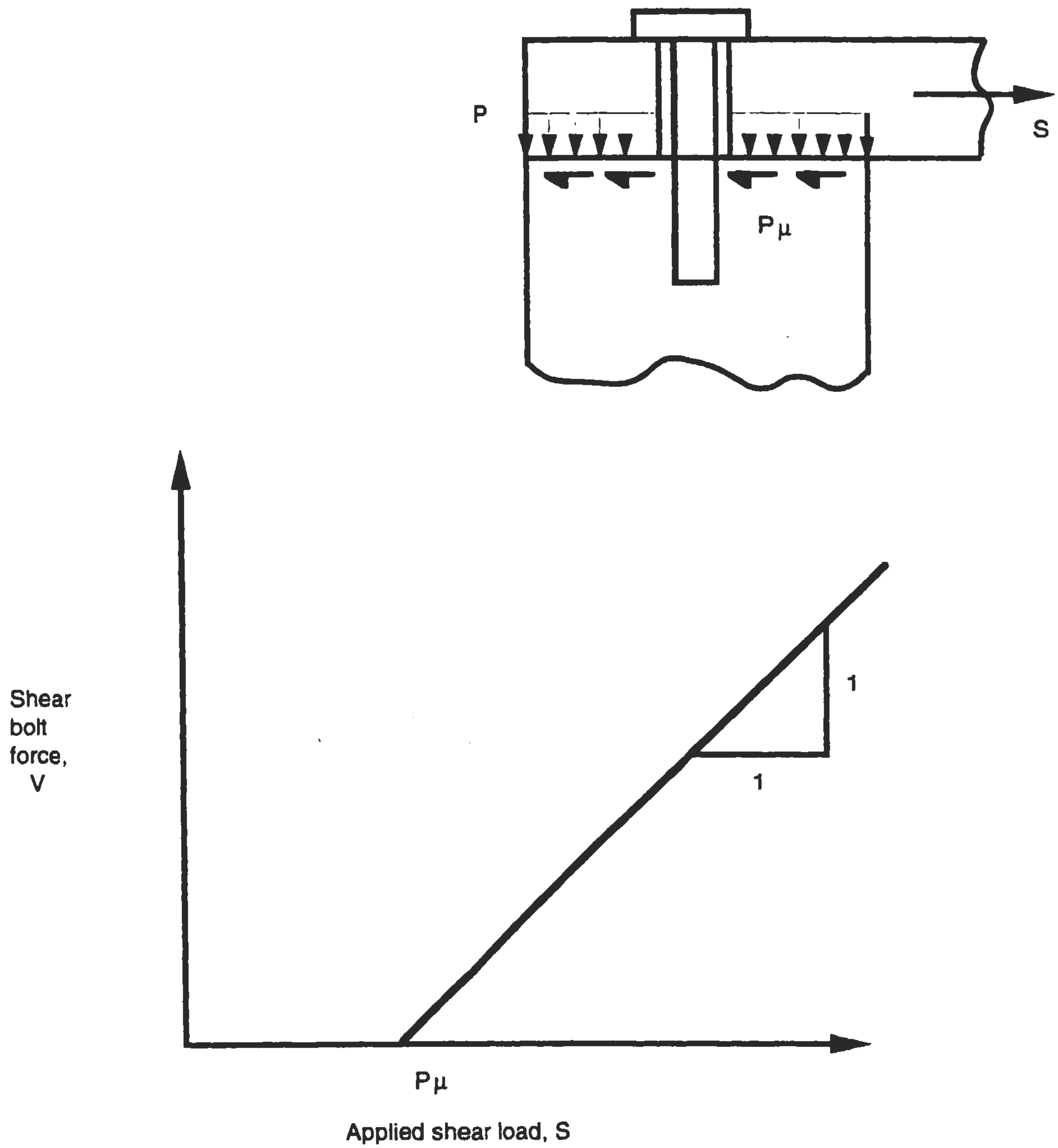
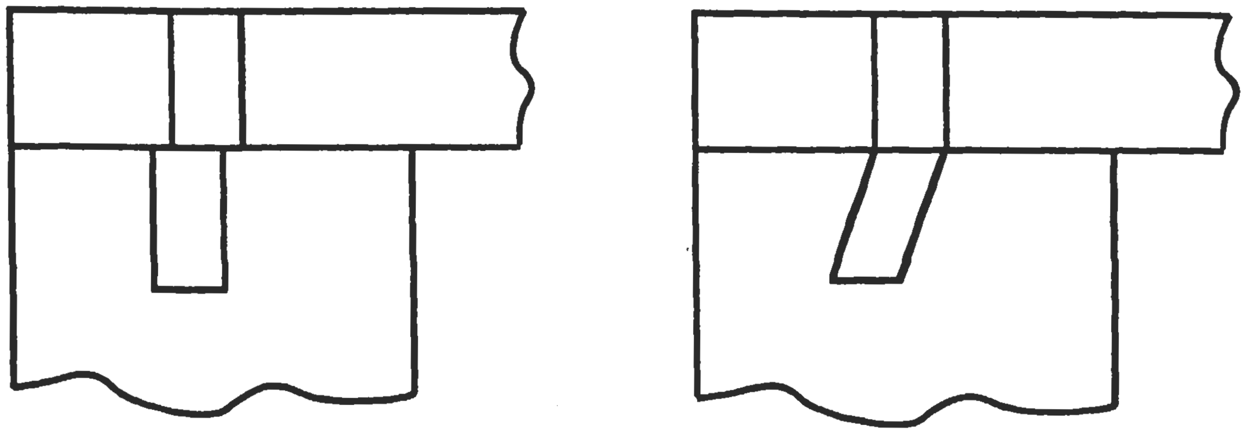
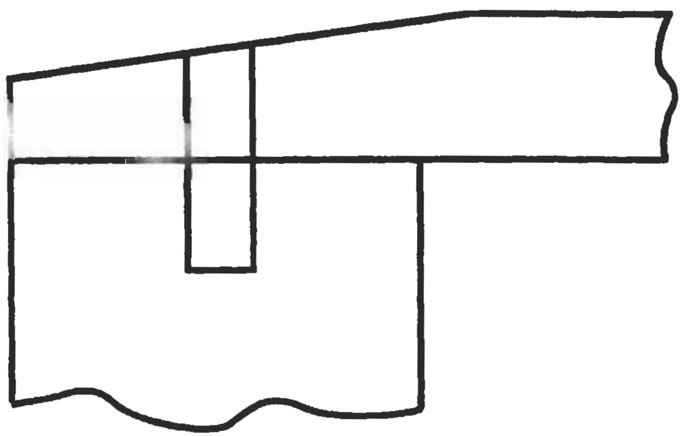


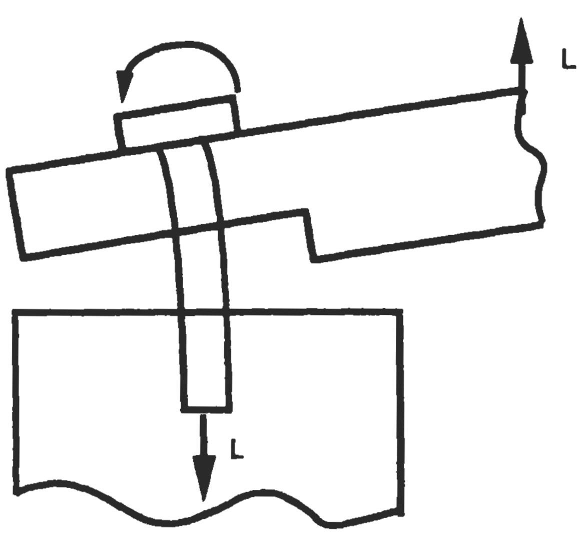
Figure I.11 The relation between shear bolt force and applied shear load.



(a) Misalignment of bolt holes



(b) Non-parallel closure surfaces



(c) Eccentric applied load

Figure I.12 Common causes for bending moment in closure bolts.

APPENDIX II

ASME Section III, Subsection NB, Design Analysis Requirements for Bolting of Class 1 Components

The ASME subsection specifies analysis requirements for all loading conditions which includes design, test, and service loading conditions. The service conditions are further subdivided into Level A, B, C, and D conditions. The Level A conditions correspond to the normal conditions in shipping casks, and the Level D to hypothetical accident conditions. The ASME requirement for Level A and D service loadings is tabulated in Tables II.1 through II.3 for comparison with the requirement for shipping casks specified in Subsection 6.1 of this report.

Table II.1 Part I, Service Loadings (Level A), Maximum Stress Analysis

Loading Class	Load	Stress Analysis	Acceptance Criteria		Code Section for Details
Service loading Level A	Combined actual service loads including preload, pressure, and temperature loads	Limit analysis of stress over cross-section	Tension		NB3232 NB3232.1 NB3233 NB3234
			Average stress	< 2 Sm*	
			Tension plus bending		NB3232.2 NB3232.3(b)
			For bolts having min. tensile strength (Su) less than 100 ksi Maximum tensile stress	< 3 Sm	
			For bolts having min. tensile strength (Su) less than 100 ksi Maximum tensile stress	< 2.7 Sm	
			Tension plus bending plus residual torsion		NB3232.2
			For bolts having min. tensile strength (Su) less than 100 ksi Maximum stress intensity	< 3 Sm	
			For bolts having min. tensile strength (Su) less than 100 ksi Maximum stress intensity	< 2.7 Sm	

*The basic allowable stress (Sm) in this table is different from the one in Table 6.1 of this report. The Sm of this table is one-half of the Sm in Table 6.1.

Table II.2 Part II, Service Loadings (Level A), Fatigue Stress Analysis

Loading Class	Loads	Stress Analysis	Acceptance Criteria	Code Section for Details
Service loading Level A	Combined actual service load histories incl. preload, pressure, and temperature loads	Fatigue analysis of stress at a point	Required unless the bolts are on a component that meet all conditions of NB3222.4(d) and thus require no fatigue analysis	NB3232.3 NB3222.4(d)
			Maximum cumulative usage factor (U) due to alternating stress intensity NB3216 < 1.0	NB3232.3(e)
			For bolts having minimum yield strength less than 100 ksi Use fatigue curves I-9.0 with elastic-modulus adjustment Use fatigue strength reduction factor not less than 4.0, unless it can be shown otherwise	NB3232.3(a) Appendix I Table I-9.0 NB3232.3(c) NB3232.3(d)
			For bolts with minimum yield strength greater than 100 ksi Use fatigue curve I-9.4 with elastic-modulus adjustment Use fatigue strength reduction factor not less than 4 Thread shall be Vee-type having minimum root radius no less than 0.003 in Fillet radius at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060	NB3232.3(b) Appendix I Table I-9.4 NB3232.3(c) NB3232.3(d)

Table II.3 Part III, Service Loadings (Level D), Maximum Stress Analysis

Load Class	Loads	Stress Analysis	Acceptance Criteria			Code Section for Details
Service load Level D	Combined actual service loads including preload, pressure, and temperature loads Also consider prying	Limit analysis of stress over cross-section	Tension			NB3235 Appendix F
			Average stress	<	Min (0.7 Su, Sy) at temperature	F1335.1
			Shear			F1335.2
			Average stress	<	Min (0.42 Su, 0.6 Sy) at temperature	
			Tension plus shear			F1335.3
			Stress ratio = computed average stress / allowable average stress Rt : Stress ratio for tensile stress Rs : Stress ratio for shear stress $Rt^2 + Rs^2 \geq 1$			
			Tension plus bending			F1335.1
			No requirement for bolts having tensile stress less than 100 ksi			
			For bolts having tensile strength (Su) greater than 100 ksi Maximum tensile stress	<	Su	

APPENDIX III

Maximum Prying Tensile Bolt Force Generated by Applied Load

1.0 Introduction

Subsection 2.2 and Appendix I of this report have demonstrated that the bending of the closure lid under an applied load is likely to produce a prying tensile bolt force and a bending bolt moment. It has also been mentioned that by using the finite element analyses reported in this appendix and Appendix IV, the prying and bending effects can be analyzed separately. Conservative estimates can be obtained from separate and simplified analyses of the prying tensile bolt force and the bending bolt moment. This appendix deals with these analyses for the prying bolt force and Appendix IV considers similar analyses for the bending moment.

The cause of the prying action in bolted closures is detailed in Appendix I. The action is simply the result of a rotation of the closure lid at the bolted joint relative to the cask wall. The occurrence of this action is controlled by the applied load, the bolt preload, as well as the local and global deformations of the closure lid, cask wall, and closure bolts caused by these loads. An evaluation of the additional tensile bolt force due to the prying action can be carried out by using the finite element method. However, the process is quite involved and expensive. A simplified approximate method with a closed-form solution is extremely desirable for design purposes. Therefore, this appendix attempts to establish such an approach to this problem and to derive the necessary formulas which a structural analyst can easily use to perform a quick assessment of the possible prying effect in a bolted cask closure. To establish the validity of this approach, the method is first described and applied to the case of a bolted tee connection which has published test results to compare with. After the present analysis results are shown to compare favorably with the test results, the method is applied to the case of a bolted closure and the formulas for the calculation of prying tensile bolt force are derived.

Two simplified models are developed and analyzed for the bolted closure. The formulas for the simpler and also the more conservative of these two models are given in Table 2.1 of this report for the calculation of the prying tensile bolt force. The formulas of both simplified models are verified with finite element analysis results based on similar assumptions. In addition, using finite element models with increasing realism, the possible effects of various assumptions and approximations used in the simplified models on the prying bolt force are studied. The study shows that the bolt bending and the cask wall flexibility (both of which are ignored in the present analysis models) do not appreciably lower the prying bolt force. Thus, the present simplified analysis methods are adequate for prying analysis. The study also shows that the bending of the closure lid under load produces more significant prying tensile bolt force than the bending bolt moment. The inability to produce significant bending moment is due to the inefficient transmission of rotation between the closure lid and the closure bolts. The transmission of rotation through a bolted joint and the significance of bending moment in closure bolt design are discussed in Appendix IV in which a simplified model is also developed for the analysis of bending bolt moment.

2.0 Bolted Tee Connections

2.1 Analysis

One of the bolted joints frequently used for the study and demonstration of the prying effect on bolt force is a bolted tee connection. Figure III.1 shows two typical connections. The first one has two identical tees whose flanges are bolted together with two identical rows of uniformly distributed bolts and nuts which are located equidistantly from the web of the tees. The second connection is identical to the first one except that one of the two tees is replaced by a semi-infinite rigid structure. When a tensile load is applied to these connections as shown in Fig. III.1, the tees and bolts in these two connections will deform in approximately the same way. Thus, the present analysis is applicable to both tee connections. Because of the symmetries of the load and geometry with respect to the x and y axes, the present analysis model includes only a quadrant of the connection. In addition, assuming uniform behavior of the connection in the length or depth direction of the tee and ignoring the end effect, the model represents only a typical unit length of the tee connection (i.e., all the variables in the model including the bolt forces and area refer to a unit length of the tee connection). To convert the bolt quantities from the per-unit length to the per-bolt basis, the results from the present analysis must be multiplied with a conversion factor C as shown below:

$$C = \frac{w}{Nb} \tag{III.1}$$

where w is the total length of the tee connection and Nb is the total number of bolts on one side of the tee connection.

Figure III.2 depicts the present analysis model in which the web of the tee is not represented because it makes an insignificant contribution to the deformation of the bolted joint. Only the tee flange and the bolt are represented as flexible members—the flange is modelled as a beam which can resist both axial and bending loads, and the bolt is modelled as a spring which can resist only axial tensile load. To make the analysis applicable to a wide range of load distributions, the applied tensile load (L) located at the center of the connection is replaced with a generic set of equivalent force and moment (Ff and Mf) located at the bolt location. The replacement force and moment must maintain equilibrium with the applied load and produce the same displacement and rotation of the tee flange at the bolt location as the applied load (L), because this displacement and rotation controls the prying action in the bolted joint. Since the fixed-end force and moment meet these two requirements for the replacement force and moment, they can be used. The fixed-end force and moment are the reaction force and moment which the applied load could generate at the bolt location if the tee flange were completely fixed at the bolt location. This reaction force and moment can be reversed in direction and used as the replacement force and moment.

Formulas are available from handbooks for determining the fixed-end force and moment for many simple loads and structures. Using information from Roark (Ref. III.1), the magnitude of the replacement force and moment for the present case are obtained for a unit length of the flange as follows:

$$Ff = L \tag{III.2}$$

$$Mf = \frac{L\,b}{2} \tag{III.3}$$

where b is the distance between the web and bolt centers. The replacement force and moment will not produce the exact deformation of the tee flange between the web and the bolt because obviously, this deformation depends also on the distribution of the applied load between the web and the bolt. However, for prying analysis of the bolted joint, the exact distribution of the deformation outside the joint has no effect on the joint behavior.

As shown in Fig. III.2, when the tensile load is applied to the center of the tee connection in the y direction, the load produces a separation and a rotation of the flange at the bolt location. The separation will reduce the joint compression existing between the contacting faces of the two connected tees—the rotation, on the other hand, will increase the compression. The increase of the joint compression by the rotation will, in turn, produce a corresponding increase in the bolt force. This bolt force increment, which is sometimes referred to as the additional bolt force due to prying, is identified in Fig. III.2 as the force R. In the same figure, the portion of bolt force that is not caused by the prying action is identified as B. Thus, the total bolt force is equal to B plus R. To balance this bolt force and the applied tensile load (Ff), the resultant joint-compression force must have the magnitude of B + R - Ff as depicted in Fig. III.2.

The non-prying portion of bolt force B can be determined using the knowledge given in Subsection 4.1 of Appendix I. It is shown there that in the absence of prying, the bolt force is approximately equal to the bolt preload (P), when the applied load (Ff) is below the bolt preload (P) and is equal to the applied load when the load exceeds the bolt preload as follows:

$$B = P \quad \text{when} \quad Ff \leq P \qquad \text{(III.4)}$$

$$B = Ff \quad \text{when} \quad Ff > P \qquad \text{(III.5)}$$

Since both the bolt preload and the applied load are given, the bolt force B is a known quantity and only the bolt force R needs to be determined. To accomplish this task, the model given in Fig. III.2 is analyzed for the deformation of the bolt and the tee under the equivalent applied load Ff and Mf.

The solution to the present problem has three regimes which are shown in Fig. III.3 as Regimes I, II, and III. The regimes are determined by the joint separation. In Regime I, the joint is only partially open. In Regime II, the joint is completely open except at the very outer edge of the tee flange. In Regime III, the joint is completely open (i.e., the tee flange is completely out of direct contact with the other tee of the connection). Formulas from Roark (Ref. III.1) for the determination of beam deflection and moment distribution under various load and boundary conditions are used to obtain the solution for each of these regimes. The right-hand side of Fig. III.3 depicts the beam cases or sub-models that are used to build the solution for the regime that is shown on the left-hand side of the same figure.

The solution for Regime III is the simplest—the additional bolt force due to prying vanishes:

$$R = 0 \qquad \text{(III.6)}$$

The condition for this regime to occur is that the applied load produces a greater bolt elongation than the deflection of the tee flange at the flange edge. The bolt elongation is the extension of the bolt after the preload has been applied—it is caused by the difference between the non-prying bolt force B and the preload P (i.e., B–P). The deflection of the tee flange is produced only by the rotation of the flange at the bolt location θ since no other load exists between the bolt and the flange edge. The mathematical condition for the occurrence of Regime III is, therefore, written as follows:

$$\frac{(B-P)Lb}{Ab\,Eb} > \theta\, a \qquad \text{(III.7)}$$

where Ab, Eb, and Lb are the bolt cross-sectional area per unit length of the tee connection, the bolt Young's modulus, and the bolt stressed length respectively, a is the distance between the bolt center and the flange edge. The rotation θ is obtained using the beam cases shown in Fig. III.3:

$$\theta = \frac{Mf\, b}{E\, I} \tag{III.8}$$

where Mf is the equivalent moment given in Equation III.3—b is the distance between the web and bolt centers, E and I are the tee-material Young's modulus, and the moment of inertia per unit flange length of the flange cross-section about its center axis:

$$I = \frac{t^3}{12} \tag{III.9}$$

where t is the thickness of the tee flange.

Equations III.7, III.8, and III.9 can be combined into one condition for the occurrence of prying as follows:

$$Mf > C2\, a\,(B - P) \tag{III.10}$$

where B is related to Ff and P as specified in Equations III.2, III.4, and III.5 and

$$C2 = \frac{E\, I}{a^2 b}\, \frac{Lb}{Ab\, Eb} \tag{III.11}$$

The solution for Regime II is more involved than Regime III because the flange edge now touches its neighbor and receives a reaction force. This reaction affects the deflection and rotation of the flange at its edge and at the bolt location. Accordingly, the results of two other beam cases or sub-models which are shown on the right-hand side of Fig. III.3 as sub-models 2 and 3 for Regime II must be included in the analysis. For this regime of the solution, it is assumed that the contact area does not spread and the location of the reaction force remains at the flange edge. Therefore, the flange deflection at the edge must be equal to the elongation of the bolt. Using formulas from Roark (Reference III.1) for the beam sub-models shown in Fig. III.3, an equation can be set up based on this condition and solved for R, the additional bolt force due to prying as follows:

$$R = \frac{\frac{Mf}{a} - C1\,(B - Ff) - C2\,(B - P)}{C1 + C2} \tag{III.12}$$

where

$$C1 = 1 + \frac{1}{3}\,\frac{a}{b} \tag{III.13}$$

and C2 is defined by Equation III.11.

This solution remains valid as long as the joint is fully open and the contact is confined to the very edge of the flange. Once the contact area starts to spread or the joint starts to close, Regime II of the solution ends and Regime I starts. The condition separating these two regimes of the solution is that the slope of the tee flange becomes zero at the edge (i.e., the flange surface becomes parallel or tangent to the joint interface). As the joint continues to close, the contact area will spread from the flange edge towards the joint center and the slope of the flange surface becomes zero over the entire contact area and boundary. The result of the contact reaction will be relocated from the flange edge to a location somewhere inside the contact area which is determined by the distribution of the contact pressure. Since it is difficult to determine the pressure distribution, the contact force for the present analysis is simply located where the slope of the flange surface is equal to zero. This boundary condition is shown in the beam sub-models of Fig. III.3 for Regime I. The distance between the contact point and the bolt (hc) is an unknown as R is. These two unknowns can be found using two conditions (namely, at the contact point the slope of the flange is equal to zero and the deflection is equal to the elongation of the bolt). Using Roark's formulas (Ref. III.1) for the sub-models shown in Fig. III.3, the following algebraic equation is obtained for the determination of hc:

$$\frac{Ab\,Eb\,Mf}{6\,E\,I\,Lb}\,hc^3 + \frac{(P - Ff)}{2b}\,(hc^2 + 2\,b\,hc) - Mf = 0 \qquad \text{(III.14)}$$

After hc is found, R is obtained using Equation III.12 and setting the variable a equal to hc.

2.2 Comparison of Analysis and Test Results

The adequacy of the foregoing analysis model and the procedure for the evaluation of prying effect can be demonstrated by comparing the analytical results to published experimental results. Table III.1 shows such a comparison. The test results were obtained by Nair et al. (Ref. III.2) using tee-connection specimens with various flange widths. By changing the distances between the web and bolt centers (b) and between the bolt center and the flange edge (a), various degrees of prying action were achieved. Table III.1 shows that for all the specimens, the present analysis compares well with the test in the results of the prying bolt force (R), which are given in the table as fractions of the applied tensile load. In the table, the solution regime of the analytical result is also identified. This information indicates that under the specified test load the bolted joint is totally separated in Specimen T3 and is touching only at the flange edge in all the other specimens.

In Figs. III.4 and III.5, the analysis and test results are further compared over the entire range of applied loads and bolt preloads used in the tests. The applied load vs. the bolt force curves of the specimens with the least and the most prying effect (i.e., T3 and T4) are shown in Figs. III.4 and III.5, respectively. For both specimens, the present analytical approach appears to be able to produce results comparable to the test results for all bolt preloads. As indicated in the results presented, Regime I of the solution prevails at low applied loads and high preloads. For a given preload, the solution changes from Regime I to Regime II as the applied load increases. At higher loads not reached by the presented results, the solution may change into Regime III especially for the cases with less prying effect and lower preload.

At higher applied loads, the analytical results of all preloads appear to converge into a single curve. A similar trend also appears to exist in the test results. However, since the test results are also affected by the yielding of the flange, it cannot be firmly concluded that the test results confirm this observation.

Figure III.5 shows that in general, the analysis results of Regime II compare closer to the test results than Regime I. This difference in the solution performance is not surprising because more assumptions are used in the derivation of the analysis formulas of Regime I than Regime II.

3.0 Bolted Closures for Shipping Casks

3.1 Analysis

The analytical approach established here for the bolted tee connection is equally applicable to the analysis of the prying effect on the closure bolt force of the shipping cask. Unfortunately, the results of the shipping cask closure are much more complicated to express in closed form than the tee connection. Using formulas for the plate and shell deformations under load and an analytical approach similar to the one described here for the tee connection, Waters and Schneider (Refs. III.3 and III.4) have derived a set of formulas for the design of bolted pipe and pressure vessel flanges in which the prying effect can be significant. The Waters-Schneider design procedure and formulas were later incorporated into the ASME Boiler and Pressure Vessel Code as Appendix L of Section III and Appendix Y of Section VIII for the analysis of bolted joints with flat-face (FF) flanges. However, the procedure had to be classified as a non-mandatory appendix because it was not well understood by design engineers and the formulas were not simple to evaluate by hand. It is, therefore, not the intent here to develop a similar procedure for the analysis of closure bolts. Instead, the primary goal is to establish a simple method which an analyst can easily use to make a quick and conservative assessment of the possible effect of prying on the closure bolt design. If the effect is shown to be significant and a more precise evaluation is desired, a detailed confirmation analysis using a computer is always possible.

Figure III.6 shows two analytical models to be used here for the analysis of prying effect in a circular bolted cask closure. The first model, using all plates to represent the closure lid, is equivalent to the beam model used for the tee-connection analysis. The second model, using a mixture of plates and rings to represent the closure lid, is crude but is easier to analyze and usually produces conservative results. Both models divide the closure lid into two areas; the circular area in the center which fits into the cask cavity, and the annular flange area which sits on the top of the cask wall. The lid thickness is uniform within each area but can be different between the two areas. Both models represent the central area of the lid as a circular plate but differ in the treatment of the flange area. For this area, the first model considers the lid to be an annular plate but the second model treats it as a circular ring. The ring treatment enables the results of the second model or the plate-ring model to be expressed in a significantly simpler form than the first model or the plate-plate model. Numerical results obtained using the two models for a typical rail cask design show that the plate-ring model usually produces more conservative results than the plate-plate model.

Because of axisymmetry, the analysis of both models is carried out for a typical pie segment of the closure lid corresponding to a unit length of the bolt circle (the circle on which the bolts are located). All the variables in the model including the bolt forces and area, thus refer to a unit length of the bolt circle. To convert the bolt quantities from the per-unit length basis used in this analysis to the per-bolt basis used in bolt design, the present results must be multiplied with a conversion factor C as follows:

$$C = \frac{2\,\pi\,b}{Nb} \qquad \text{(III.15)}$$

where b is the radius of the bolt circle and Nb is the total number of bolts on the bolt circle.

The approach and assumption used for the analysis of the cask closure are essentially the same as for the tee connection which are described in Subsection 2.1 of this appendix. The details are not repeated here. Similar to the tee connection, the solution of the cask closure also has three regimes. However, only the solution for Regime II is needed for evaluating the worst prying bolt force because the most critical prying condition occurs in Regime II and near the transition from

Regime II to Regime III. Near the transition, the bolted joint is sufficiently open to lose most of the help from the bolt preload and the joint compression in resisting the prying action, but the joint is not sufficiently open to eliminate the prying action. Thus, the worst prying bolt force is likely to occur near the transition. This expectation is confirmed in the studies presented later in Section 3.2 of this appendix.

Similar to the tee connection, the applied load on the bolted closure is represented by an equivalent force and moment at the bolt circle, i.e., Ff and Mf, respectively. Figure III.6 shows these equivalent loads and the supporting bolt and reaction forces. A comparison of this figure to Fig. III.2 confirms that the same generic loads and forces are involved in the closure and tee-connection analyses. Roark's (Ref. III.1) formulas for circular and annular plates of uniform thickness are used to determine the equivalent loads from the applied load and to obtain the deflection and rotation of the closure lid. However, the closure analysis is more complicated than the tee connection because the formulas are more complicated and more submodels must be used to accommodate the two different thicknesses in the closure lid. The additional sub-models introduce into the analysis, unknowns which are the internal bending moments depicted in Fig. III.7 at the interface of two adjacent sub-models. Additional equations must be formed on the basis of continuous rotation and balanced moment across the boundary of adjacent submodels. These additional equations are solved for the internal moments in terms of the applied loads (Ff and Mf) and the bolt forces (B, R, and P), so that the lid rotation at the bolt circle and the lid deflection at its edge can be expressed as functions of the applied loads and forces only. Once these functions are established, the prying condition and the prying bolt force can be determined in terms of the applied load and bolt preload in exactly the same manner as the tee connection.

To further demonstrate the similarity between the solutions of the bolted closure and tee connection, the final results of the closure lid are presented here in the same general form as Inequality III.10 and Equation III.12 for the tee connection. The plate-plate model of the bolted closure predicts the occurrence of prying if the following condition exists:

$$Mf > C2\,(c - b)\,(B - P) \qquad \text{(III.16)}$$

where

$$C2 = \frac{1}{(c-b)K1}\,\frac{1}{c^2}\,\frac{E\,tf^3}{12(1-\nu^2)}\,\frac{Lb}{Ab\,Eb} \qquad \text{(III.17)}$$

Under this condition, the prying bolt force is determined by the applied load and the bolt preload as follows:

$$R = \frac{\dfrac{Mf}{c-b} - C1\,(B - Ff) - C2\,(B - P)}{C1 + C2} \qquad \text{(III.18)}$$

where B is given by Equations III.2, III.4 and III.5; it is equal to Ff or P depending on whether Ff is greater or less than P, respectively; C2 is given by Equations III.17;

$$C1 = \frac{c}{c-b}\,\frac{K2}{K1} \qquad \text{(III.19)}$$

$$K1 = (1 + MBM)\left(\frac{C1III}{C7III}\,L8IIIb - L2IIIb\right) \qquad \text{(III.20)}$$

$$L8IIa = \frac{1}{2}\left[1 + \nu + (1 - \nu)\left(\frac{a}{b}\right)^2\right] \qquad \text{(III.36)}$$

$$L8IIb = 1 \qquad \text{(III.37)}$$

$$L8IIIb = \frac{1}{2}\left[1 + \nu + (1 - \nu)\left(\frac{b}{c}\right)^2\right] \qquad \text{(III.38)}$$

$$L9IIIb = \frac{b}{c}\left\{\frac{1+\nu}{2}\ln\frac{c}{b} + \frac{1-\nu}{4}\left[1 - \left(\frac{b}{c}\right)^2\right]\right\} \qquad \text{(III.39)}$$

Comparing the expressions for the prying condition (Inequalities III.10 and III.16) and for the prying bolt force (Equations III.12 and III.18), the results of the bolted closure and of the bolted tee connection are practically identical except in the detailed definition of the coefficients C1 and C2. The same formulas Inequality III.16 and Equation III.18, also hold for the results of the plate-ring model of the bolted closure if the coefficients C1 and C2 are given as follows:

$$C1 = 1 \qquad \text{(III.40)}$$

$$C2 = \frac{1}{12\, b\, (c - b)^2}\left[\frac{El\, tl^3}{1 - \nu} + \frac{(c - a) Ef\, tf^3}{b}\right]\frac{Lb}{Ab\, Eb} \qquad \text{(III.41)}$$

where El and Ef are the material Young's modulus of the lid center and of the flange, respectively.

These expressions are significantly simpler than the corresponding expressions of the plate-plate model. This simplicity will give the plate-ring model a distinct advantage over the plate-plate model in design application if the two methods produce comparable results. Table III.2 compares the prying bolt forces obtained using these two models for several closure lid designs of a typical rail cask. The comparison shows substantial agreement between the two methods for closure lids with various thicknesses. Therefore, only the plate-ring model has been recommended in Table 2.1 of this report for the analysis of the prying effect on closure bolts.

The prying-force results in Table III.2 indicate two interesting properties concerning the prying force: (1) the magnitude of the prying force diminishes quickly with increasing closure lid thickness; and (2) the magnitude of the prying force does not show a consistent relationship with the bolt preload. The first property simply confirms the description given in Appendix I of the prying force. The second property, however, appears unreasonable at first sight, because the joint compression produced by the bolt preload should help resist the rotation of the closure lid and thus the preload should have a definite effect on the magnitude of the prying force. This expectation would be valid if the contribution of the joint compression were significant. Since the distance between the bolt and the lid edge is small, the resistance to the lid rotation offered by the joint compression cannot be significant. Accordingly, the bolt preload will have an insignificant influence on the prying force. The apparent lack of correlation of the preload and the prying force in Table III.2 does not indicate that the present analysis is invalid. The validity of the present analysis is confirmed in the next section by comparison with finite element analysis results.

Theoretically, as explained in Section 2.1 of this appendix, the equivalent force and moment (Ff and Mf) used for the prying analysis are the fixed-edge force and moment generated by the applied load at the bolt location of the actual structure. For a closure lid with two thicknesses, the determination of this force and moment is not simple. In this report we ignore the existence of two thicknesses and the formulas for the fixed-edge force and moment of a uniform plate are always used. Figure III.8 shows the formulas for three common loadings encountered in cask analysis, namely, a concentrated normal load at the plate center, a uniformly distributed pressure over the entire plate, and a linear temperature gradient through the plate thickness. The formulas for these cases can be obtained using formulas given in Ref. III.1. For the case of a concentrated load L at the center of a uniform plate of radius b, the fixed-end force and moment at the plate edge (r=b) are given as follows:

$$Ff = \frac{L}{2 \pi b} \qquad \text{(III.42)}$$

$$Mf = \frac{L}{4 \pi} \qquad \text{(III.43)}$$

The same formulas for the case of a uniform pressure (p) are as follows:

$$Ff = \frac{p\, b}{2} \qquad \text{(III.44)}$$

$$Mf = \qquad \text{(III.45)}$$

For the case of a circular plate with a linear temperature gradient,

(III.46)

$$Mf = \frac{E \alpha t^2 DT}{12 (1 - \nu)} \qquad \text{(III.47)}$$

where E, ν, α, and t are the Young's modulus, the Poisson's ratio, the coefficient of thermal expansion and the thickness of the plate, respectively.

3.2 Application and Verification of Analysis

To confirm the validity of the simplified models and formulas developed in the preceding subsection for the bolt prying force and to assess the possible extent of prying effects in actual bolted closures, the formulas are used here to obtain the prying tensile bolt force of three possible bolted closure designs for a typical rail cask. The criteria used to set the closure lid thickness and the bolt area are given in Table III.3 with the numerical results. To demonstrate the possible effects of the relative lid and bolt stiffness on the results, the designs covers both extreme situations of relative stiffness (i.e., a weak lid material matched with a strong bolt material and a strong lid material with a weak bolt material).

For each design, various preloads and pressure loads are considered. For each of the load cases, the bolt prying force is obtained using the two simplified formulas and several finite element models. All of the results are listed in Table III.3. The value of 0 for the prying force indicates that the bolted joint is either separated or that no appreciable prying force is generated. The assumptions and details of the finite element models vary and they are used to demonstrate various effects on the prying tensile bolt force. The details and purpose of all the models are given in Table III.4 and the geometry is depicted in Fig. III.9. The finite element programs used are the

GEMINI and the NIKE programs (Refs. III.5 and III.6). GEMINI is for linear elastic analysis only. NIKE can also be used for non-linear analysis. The only nonlinear capability of NIKE used here is the sliding interface—the plasticity option is not used.

The GEMINI 1 model is very similar to the simplified analysis models developed here for the analysis of prying and is intended for the verification of the results from the simplified models and formulas. Table III.3 shows that the GEMINI 1 results compare closely with those of the simplified models.

The GEMINI 2 model includes the bolt bending effect in addition to the prying effect. The difference between the results of this model and the GEMINI 1 model represents the effect of bolt bending. The comparison of the prying bolt forces from these two models in Table III.2 shows that the bolt bending has an insignificant effect on the prying bolt force. Therefore, ignoring the bolt bending in the present simplified models for prying analysis is justified.

The NIKE models (1 and 2) include the cask wall elasticity as well as other properties of the bolted joint. A comparison of the prying bolt forces obtained using the simplified models with those of the NIKE models suggests that the cask wall elasticity may have a significant effect on the prying bolt force. However, the results of the simplified models are consistently higher than the NIKE results. Therefore, the simplified models are conservative for bolt design.

The NIKE 1 model represents the interface between the bolt head and the lid as a bonded surface while the NIKE 2 model considers it to be a sliding surface. The NIKE 2 model is relatively more realistic. However, the results of Table III.3 show that the additional realism of NIKE 2 does not make an appreciable difference in the prying bolt force. As shown in Appendix IV, the same conclusion also appears to hold for the bending bolt moment. These results suggest that for the transmission of force and moment between the closure lid and the bolt, the interface between the bolt head and the lid can be considered practically bonded.

A review of all the results in Table III.3 shows that the agreement between the results of the simplified models and the finite element models is generally good for all bolted joint designs, bolt preloads, and applied loads. This general agreement over a wide range of designs, preloads, and applied loads demonstrates the adequacy of the simplified models and formulas for the evaluation of prying bolt force. The review of the results in Table III.3 produces additional observations concerning the prying effects in bolted closures. These observations and the other observations already discussed in this appendix are summarized as follows:

- A closure lid that is adequately designed to support an applied load may not have a sufficient thickness to avoid a significant prying action. Accordingly, the prying action should be checked for all designs.

- A design with a combination of a weak lid material and a strong bolt material has less prying action than a design with the combination of a strong lid material and a weak bolt material. The result is mainly due to the fact that a weaker lid material leads to a thicker lid which reduces the prying action.

- The greatest prying action occurs when the applied load is equal to the preload (i.e., when the closure lid has just completed its separation from the cask wall and the joint compression is no longer available to resist the bending of the closure lid). Therefore, the preload should never be set equal to, or close to, the most critical load.

- The prying action can be minimized by a number of means including the adjustment of bolt preload and bolt area; however, the stiffening or thickening of the closure lid produces the most predictable results. The thickening of the lid is required only for the lid area over the cask cavity. The results of Table III.2 indicate that having a smaller lid thickness over the cask wall than over the cask cavity may help reduce the prying action. The plate-plate model is developed specifically for the analysis of this case of closure lid with two different thicknesses.

- The bolt bending has insignificant effects on the prying bolt force and can be omitted in the evaluation of the bolt force. However, the cask wall elasticity may have a significant effect and can be included to reduce the conservatism of the present simplified models for the analysis of prying bolt force.

In conclusion, the additional bolt force due to prying can be significant in bolt design and the simple analysis methods and formulas developed here can be used to facilitate its evaluation.

3.3 Prying Action of Inward Load

The foregoing analysis of prying action is for an outward applied load (i.e., a load directed toward the exterior of the shipping cask). An inward applied load can also produce an additional tensile bolt force by prying although it does not produce a non-prying tensile bolt force because the applied load is supported by the cask wall not the closure bolts. Both the inward and the outward load generate the prying action by bending the closure lid. Figure III.10 compares the prying actions generated by an inward and an outward load in a shipping cask closure. The reaction force (R) which causes the prying effect on the bolt force is located at the outer edge of the closure lid in the case of the outward load, but is at the inner edge of the cask wall in the case of the inward load.

Figure III.10 also depicts the analytical models which may be used for the solution of the reaction force and the resulting bolt force. The two models are analytically the same except in the location and magnitude of the supports and forces. If the cask-cavity radius is much greater than the cask-wall thickness, the difference in load location can be ignored and the only remaining difference between the two models is in the force magnitudes. In the inward load model, the fixed-edge force Ff is directly supported by the cask wall and consequently it does not enter into the solution of the prying force as in the outward load model. In other words, the prying action of the inward load is caused only by the fixed-edge moment. Thus if the fixed-edge force is ignored or set to zero in the outward load model, the model and all its formulas developed here for prying analysis of the outward load can be readily used to obtain the prying bolt force, R for the inward load.

In the outward load case, the preceding subsection has shown that the maximum prying bolt force occurs when the applied load is equal to the preload (P); (i.e., when P = Ff.) Since the inward load solution is the same as the outward load solution with Ff set to 0, the condition for the maximum prying force to occur in the inward load case is when P = Ff = 0, (i.e., when there is no preload.) Thus, the application of a preload always helps reduce the prying tensile bolt force generated by an inward load although the same preload may enhance the prying action of an outward load.

References (Appendix III)

III.1 W. C. Young, *Roark's Formulas for Stress and Strain*, 6th Ed., McGraw Hill, New York, NY, 1989.

III.2 R. S. Nair, P. C. Birkemoe, and W. H. Munse, "High Strength Bolts Subjected to Tension, and Prying," *Journal of the Structural Division, ASCE*, Vol. 100, ST2, February 1974.

III.3 E. O. Waters and R. W. Schneider, "Axisymmetric, Nonidentical, Flat Face Flanges with Metal-to-Metal Contact Beyond the Bolt Circle," *ASME Journal of Engineering for Industry*, Vol. 91, No. 3, August 1969.

III.4 E. O. Waters, "Derivation of Code Formulas for Part B Flanges," *WRC Bulletin*, No. 166. October 1971.

III.5 R. C. Murray, "GEMINI-A Computer Program for Two and Three Dimensional Linear Static, and Seismic Structural Analysis," Lawrence Livermore National Laboratory Report UCID-20338, October 1984.

III.6 B. Engelmann, "NIKE2D-A Nonlinear, Implicit, Two-Dimensional Finite Element Code for Solid Mechanics-User Manual," Lawrence Livermore National Laboratory Report UCRL-MA-105413, April 1991.

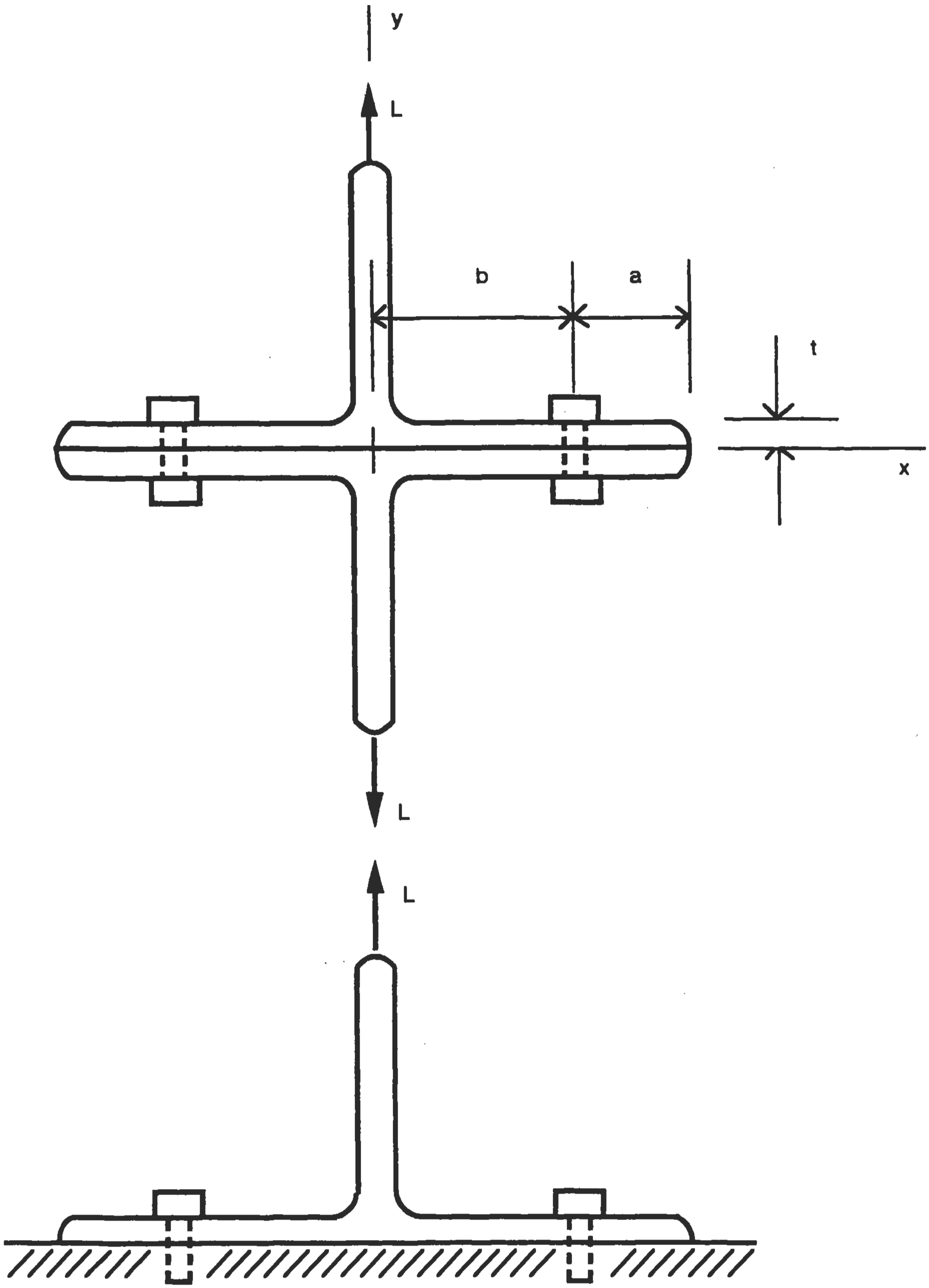


Figure III.1 Bolted tee connections and applied tensile load.

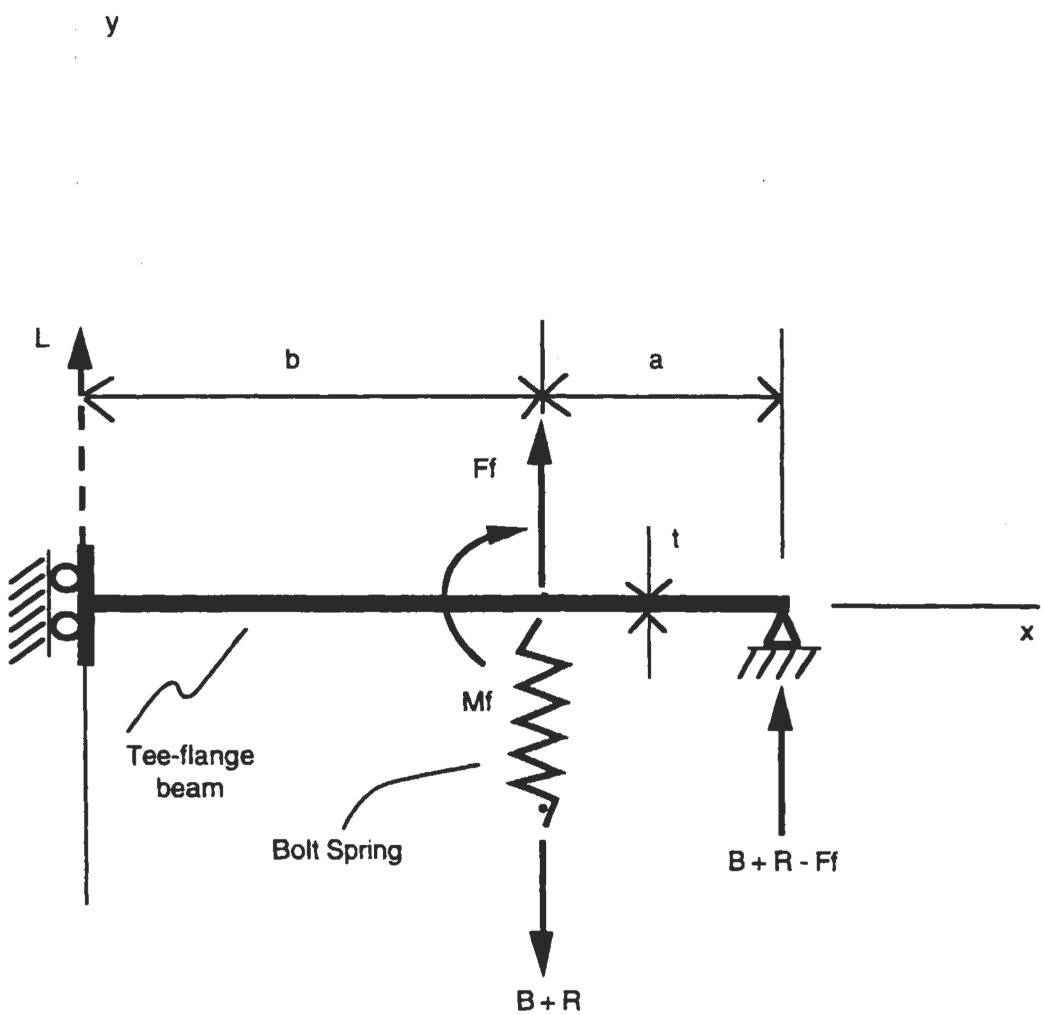


Figure III.2 Model used for the analysis of prying in tee connections.

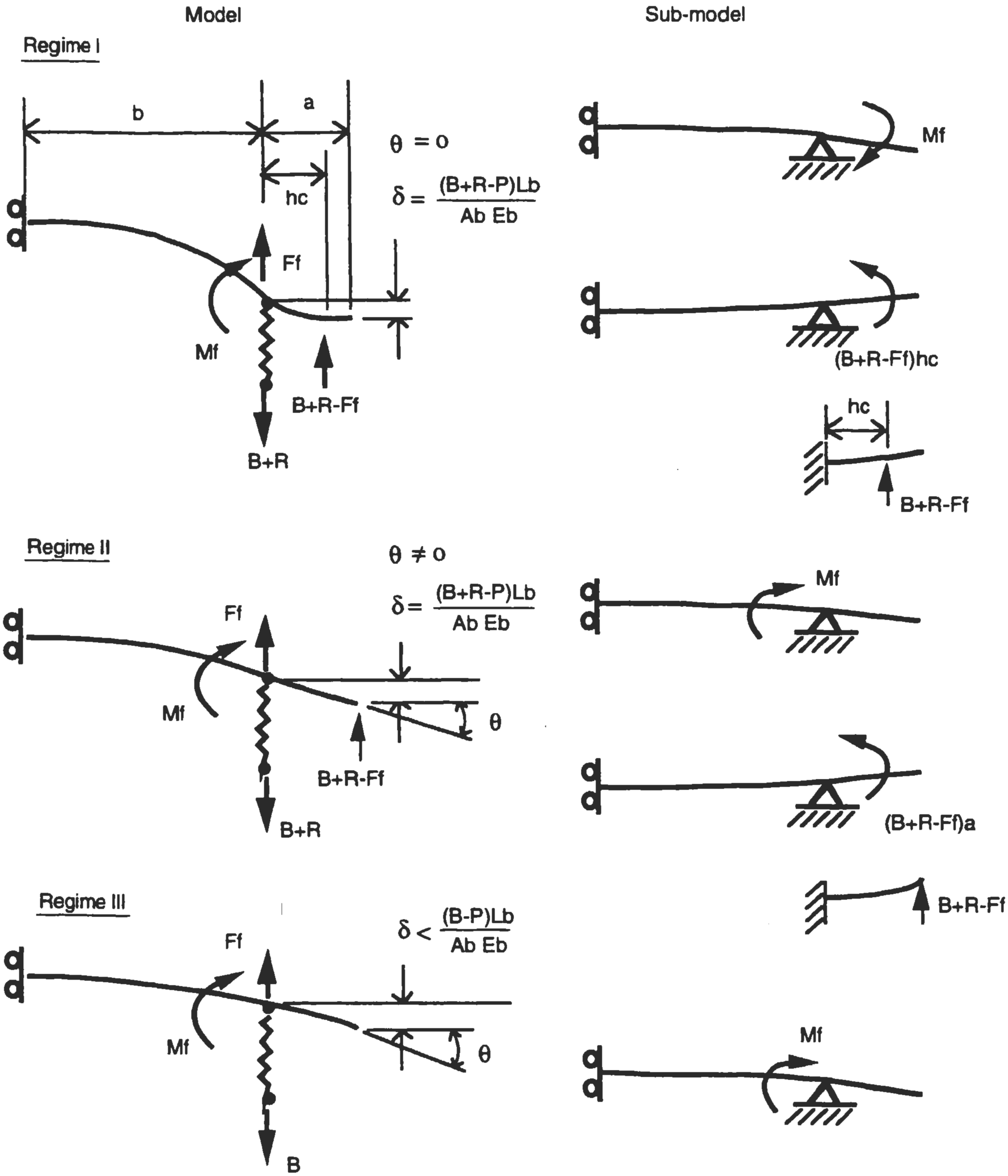


Figure III.3 Possible regimes for prying solution and corresponding sub-models for obtaining the solution.

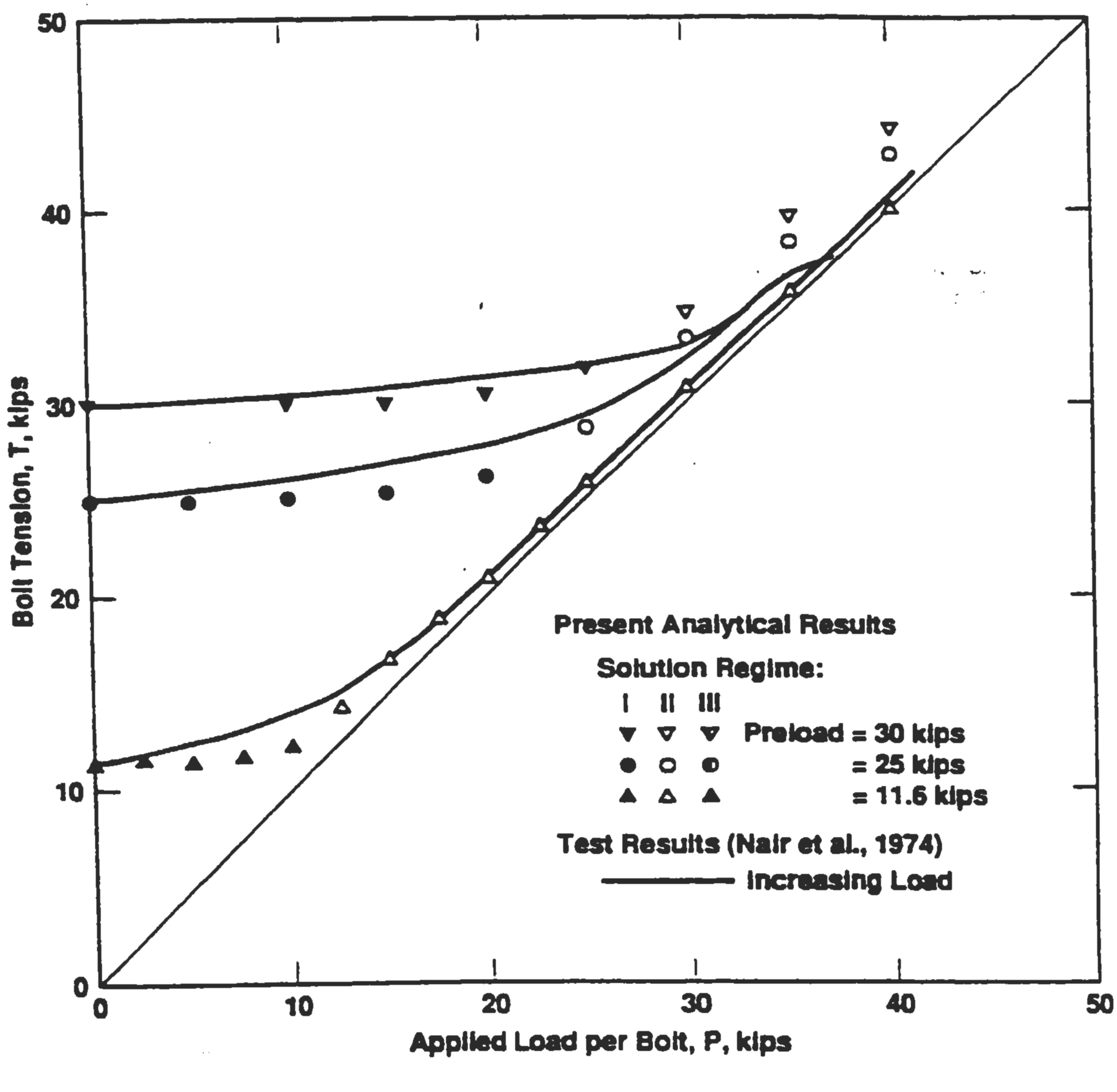


Figure III.4 Comparison of analytical and experimental results of prying bolt force for a typical case (T3) having minimal prying effect.

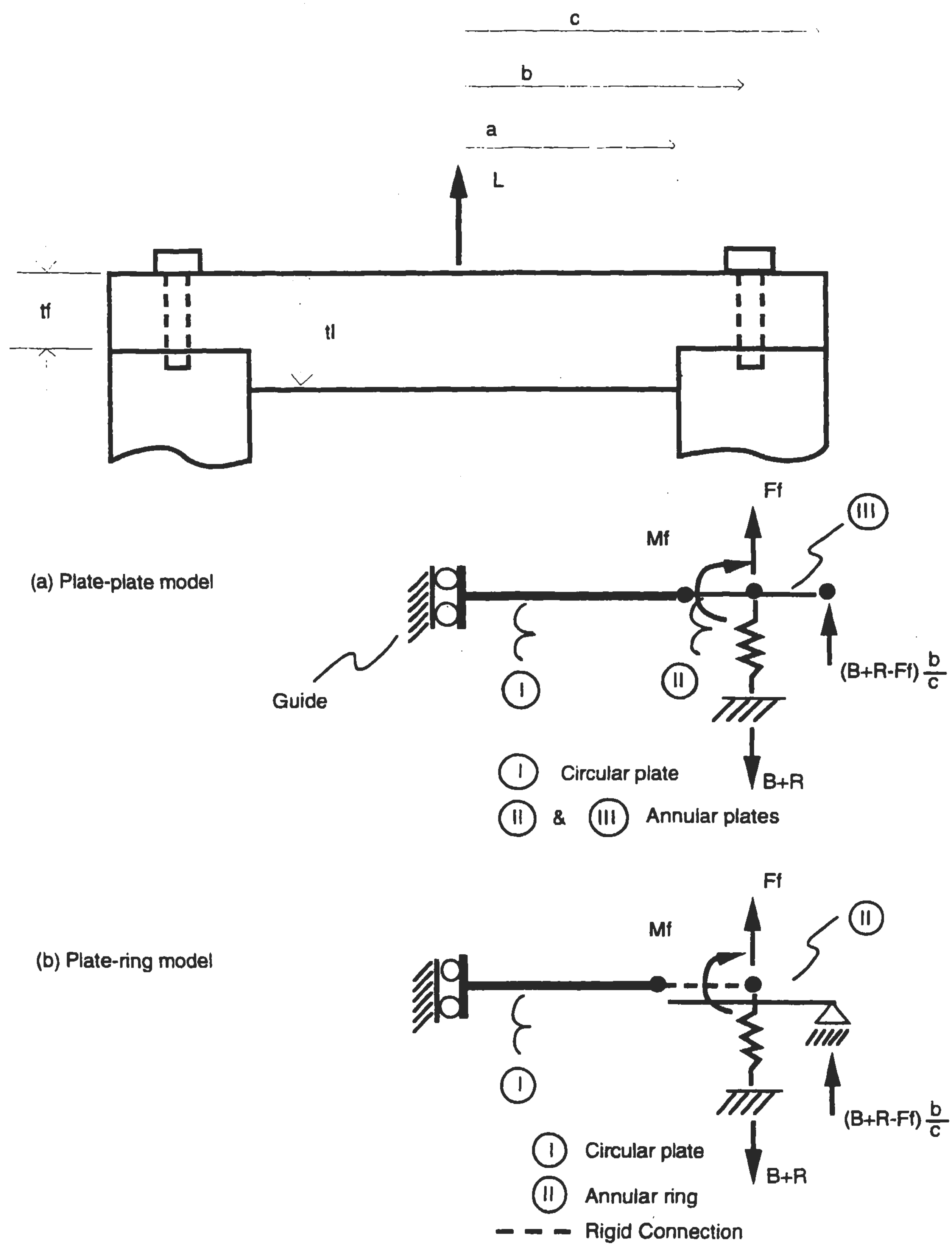
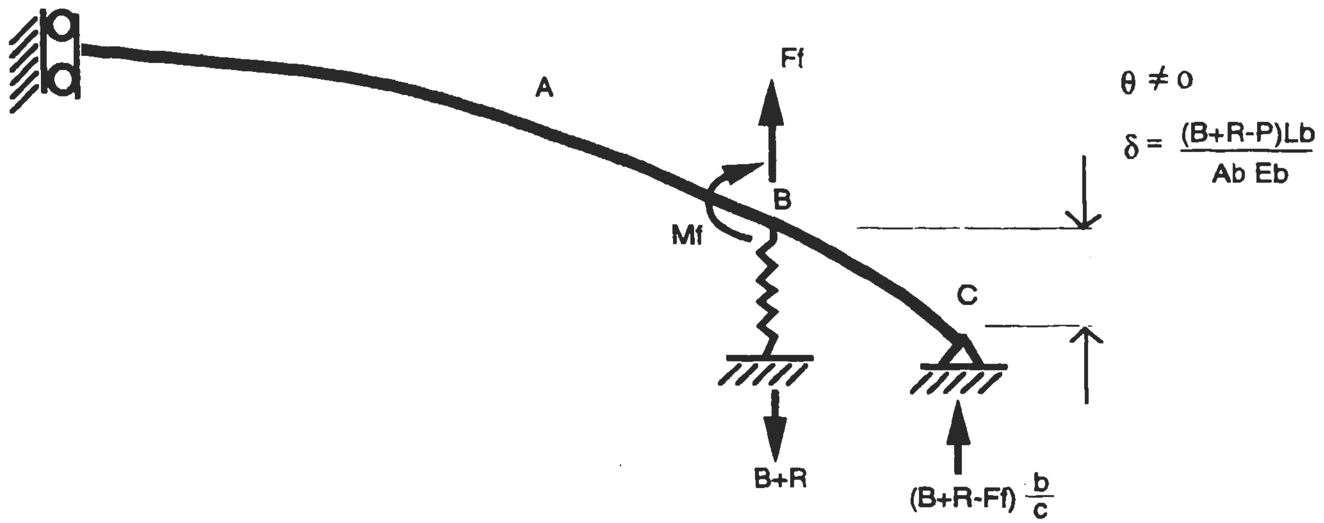
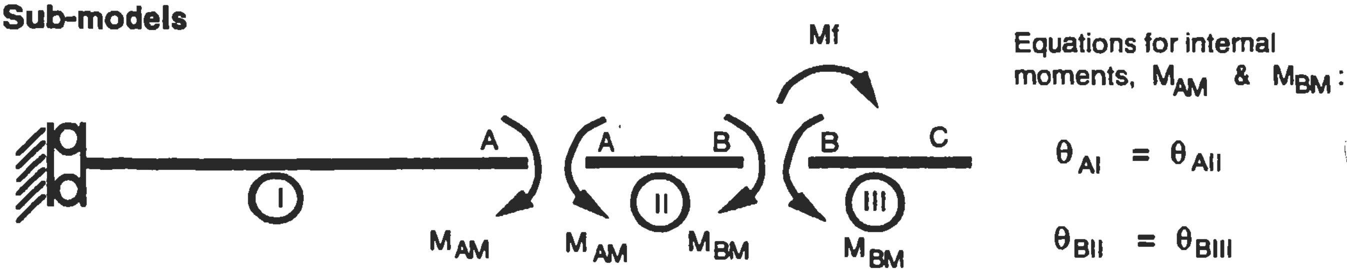


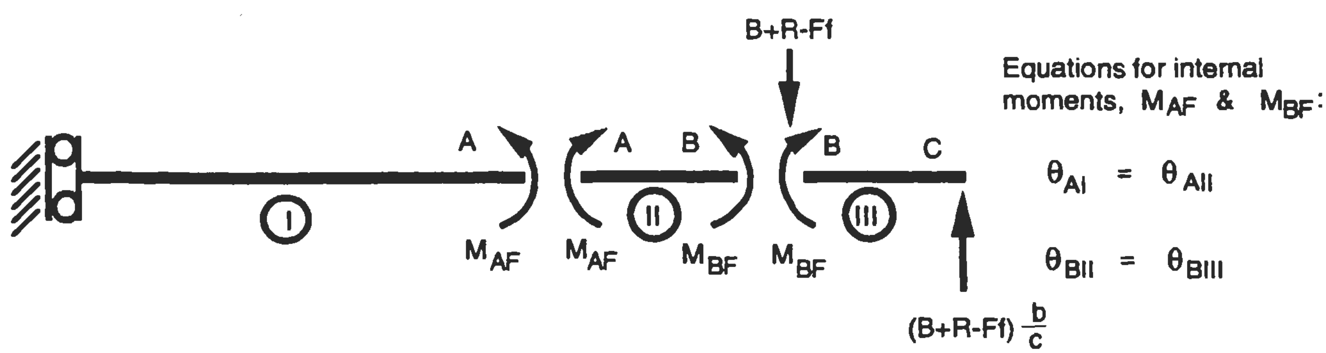
Figure III.6 Analytical models for the evaluation of prying bolt force in circular shipping casks; (a) the plate-plate model, and (b) the plate-ring model.

Model



Sub-models





$$\delta_{total} \text{ at B} = \frac{(B+R-P)Lb}{Ab\ Eb}$$

Figure III.7 Sub-models used for the analysis of the plate-plate model of a bolted closure.

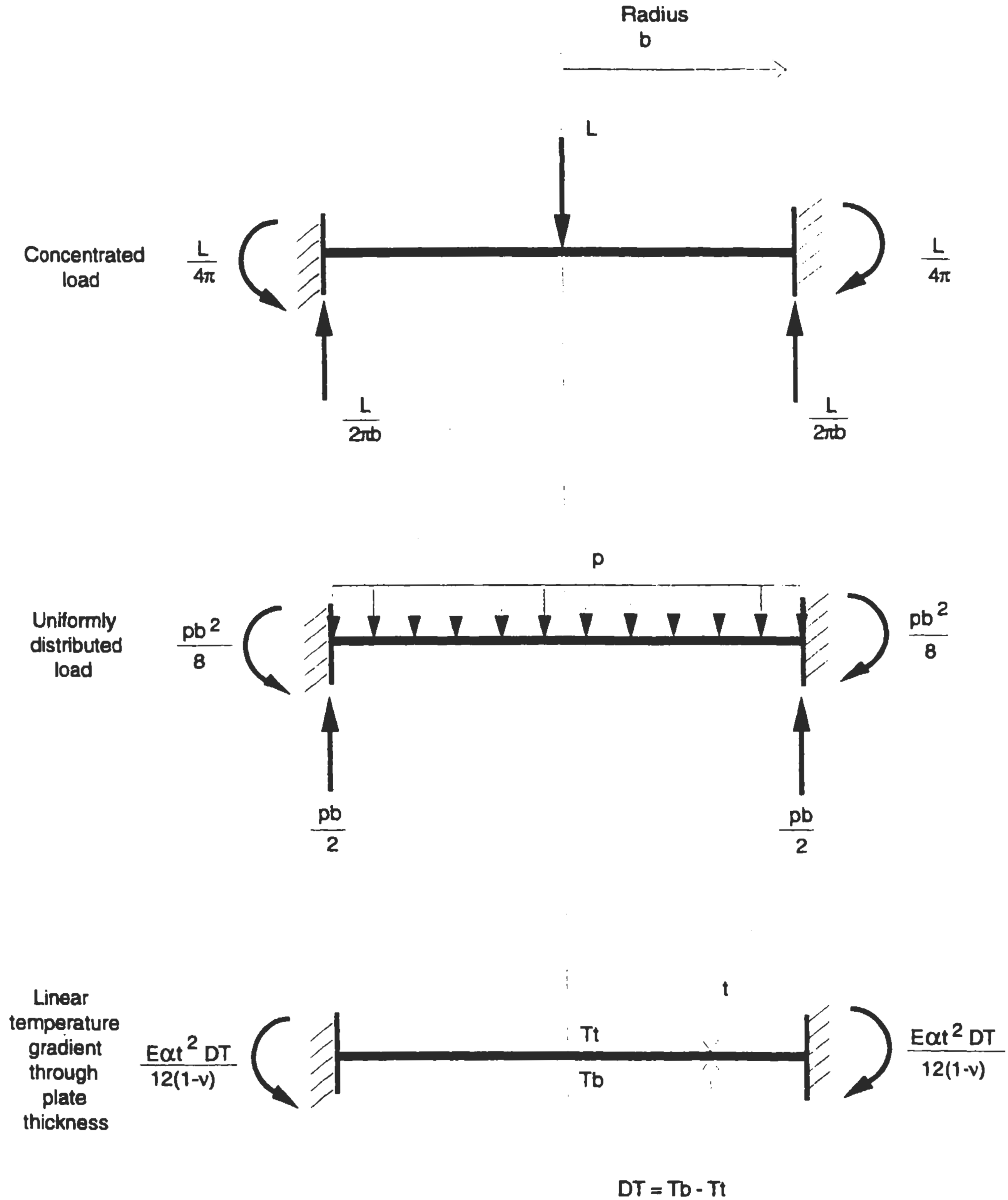


Figure III.8 Formulas for fixed-edge force and moment of a circular plate of uniform thickness.

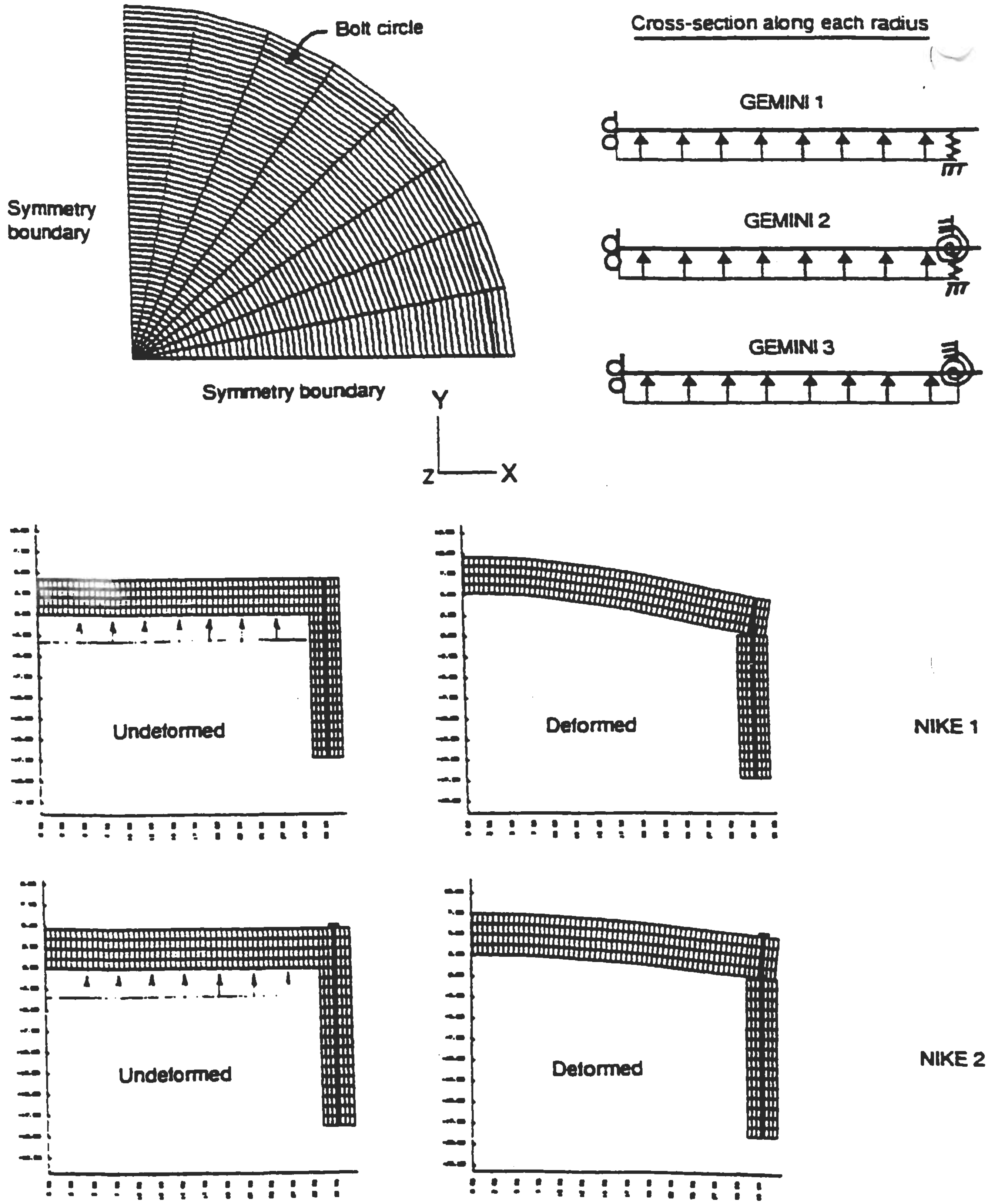
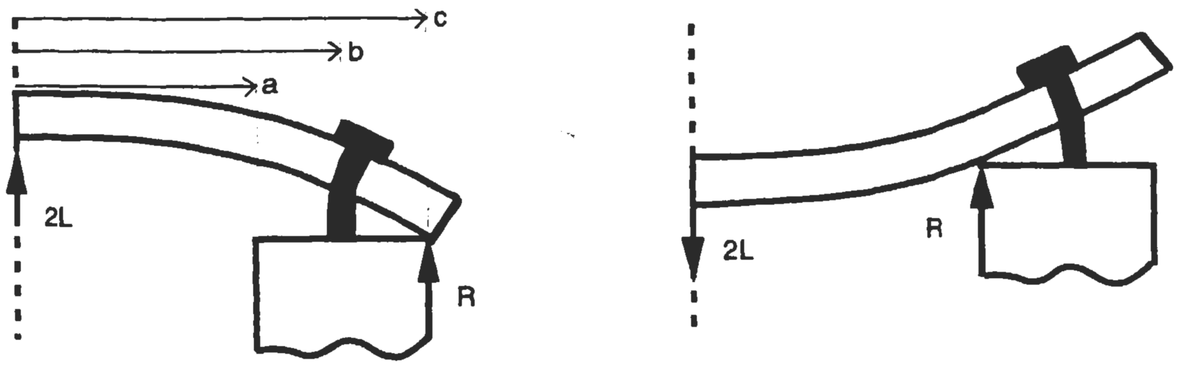


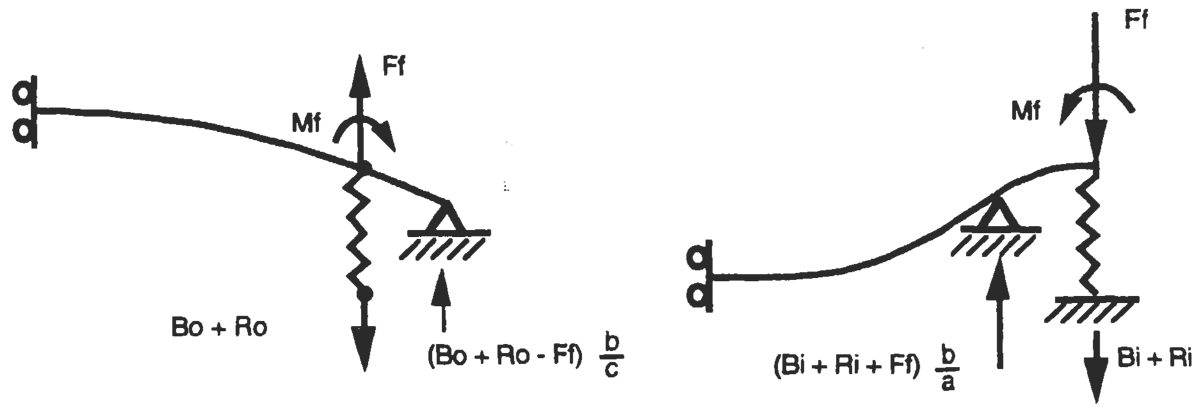
Figure III.9 Finite element models used for the evaluation of prying bolt force.

Outward Load | Inward Load

Shipping Cask



Analysis Model for Prying Tensile Bolt Force



Non-prying tensile bolt force (Bo)

When Ffo < or = P, Bo = P
When Ffo > P, Bo = Ffo

Prying tensile bolt force (Ro)

Ro

Total tensile bolt force (Bo + Ro)

When Ffo < or = P, Bo + Ro = P + Ro
When Ffo > P, Bo + Ro = Ffo + Ro

Non-prying tensile bolt force (Bi)

For all Ffi, Bi = P

Prying tensile bolt force (Ri)

Ri

Total tensile bolt force (Bi + Ri)

For all Ffi, Bi + Ri = P + Ri

Difference and Similarity between Outward- and Inward- Load Analysis Models

Although there are differences in load location, the main difference between the two models is in the role of the fixed-edge force, Ff. In the case of inward load, Ff is supported by the cask wall and thus has no effect on both the non-prying and prying bolt forces, B and R. Therefore, if the effect of Ff is ignored; i.e., if Ff is set to zero in the outward load model, the model and its formulas can be readily used for the inward load.

Figure III.10 Comparison of prying actions of inward and outward applied loads.

Table III.1 Comparison of Test and Analysis Results for the Prying Bolt Force to Applied Load Ratio (R/L) of Various Bolted Tee-connection Specimens

Bolted Tee-Connection Specimens	Specimen Dimensions						Loads		Comparison of Results		
	Flange		Bolt		Distance between				Ratio of Prying Bolt Force to Applied Load (R/L)		
	Width (per bolt)	Thickness	Nominal Diameter	Stressed Length	Web & Bolt Centers	Bolt Center & Flange Edge	Bolt Preload	Applied Tensile Load at Web Center (per bolt)	Test	Present Anal. Results	
	w (in)	t (in)	d (in)	Lb (in)	b (in)	a (in)	P (lbs)	L (lbs)	Result	Sol. Regime	Result
T0	3.75	1.06	0.75	1.69	1.75	1.72	0	152000	0.10	II	0.14
T1	3.75	1.06	0.75	1.69	1.00	1.72	0	157000	0.07	II	0.01
T2	3.75	1.06	0.75	1.69	2.50	1.72	0	174000	0.05	II	0.13
T3	3.75	1.06	0.75	1.69	1.75	1.09	0	202000	0.02	III	0.00
T4	3.75	1.06	0.75	1.69	1.75	2.59	0	142000	0.37	II	0.40

Table III.2 Comparison of Prying Bolt Forces Obtained Using the Plate-plate and the Plate-ring Models of the Closure Lid

	Closure lid thickness		Cask internal pressure	Bolt preload	Non-prying bolt force	Prying bolt force	
						Plate-ring model	Plate-plate model
	Center	Flange		per unit length of bolt circle			
Typical bolted rail cask closure dimensions & material properties	tl (in)	tf (in)	p (psi)	P (lb/in)	B (lb/in)	R (lb/in)	R (lb/in)
Cask & closure lid outer radius:	7	7	100	0	1638	0	0
c = 34.50 in	3.5	3.5	100	0	1638	4408	4464
	2	2	100	0	1638	7157	7079
Cask cavity and lid center radius:							
a = 30.00 in	7	7	100	1000	1638	226	263
	3.5	3.5	100	1000	1638	4758	4800
Typical lid thickness: 7 in	2	2	100	1000	1638	7211	7130
Other thicknesses used here are							
for demonstrating the possible	7	7	100	2000	2000	759	791
difference between the results of	3.5	3.5	100	2000	2000	4745	4773
the two analysis models	2	2	100	2000	2000	6903	6818
Bolt circle radius:	3.5	7	100	0	1638	1759	1681
b = 32.75 in	1.75	3.5	100	0	1638	6750	6740
	1	2	100	0	1638	7555	7577
Bolt nominal diameter:							
Db = 1.625 in	3.5	7	100	1000	1638	2394	2325
	1.75	3.5	100	1000	1638	6848	6842
Bolt stress length:	1	2	100	1000	1638	7566	7589
Lb = tf							
	3.5	7	100	2000	2000	2666	2607
Total number of bolts:	1.75	3.5	100	2000	2000	6584	6581
n = 36	1	2	100	2000	2000	7215	7238
Bolt area per unit length of	10.5	7	100	0	1638	0	0
bolt circle:	5.25	3.5	100	0	1638	1801	2347
Ab = 0.3628 in**2/in	3	2	100	0	1638	6231	6245
Young's modulus of closure lid:	10.5	7	100	1000	1638	0	0
E = 28000000 psi	5.25	3.5	100	1000	1638	2431	2900
	3	2	100	1000	1638	6385	6362
Young's modulus of bolt:							
Eb = 28000000 psi	10.5	7	100	2000	2000	258	332
	5.25	3.5	100	2000	2000	2699	3092
Poisson's ratio of closure lid:	3	2	100	2000	2000	6176	6116
v = 0.3							

Table III.3 Additional Tensile Bolt Force Caused by Prying in Sample Rail Cask Designs

Case 1: Lower-strength lid material (304 stainless steel, minimum yield strength = 25000 psi, tensile strength = 65000 psi)
Higher-strength bolt material (SA 540, minimum yield strength = 150000 psi, tensile strength = 165000 psi)

Closure lid thickness determined by accident load and allowable bending stress (60000 psi)
Bolt area determined by accident load and allowable tensile stress (150000 psi), no prying is considered.
Accident load, 1000 psi; Normal load, 100 psi

	Closure lid		Closure bolt								Predicted additional prying tensile bolt force per unit length of b.c.					
											Present simpl. methods		Finite element method			
											Bolt bending not included			Bolt bending included		
	Edge radius	Thick-ness	Bolt circle radius	Area/unit length of b.c.	Length	Pressure load	Fixed-edge force, Ff per unit length of b.c.	Fixed-edge moment, Mf per unit length of b.c.	Bolt preload per unit length of b.c.	Non-prying bolt force per unit length of b.c.	Plate-ring model Rigid wall	Plate-plate model Rigid wall	GEMINI model 1 Rigid wall	GEMINI model 2 Rigid wall	NIKE model 1 Flex. wall	NIKE model 2 Flex. wall
	(in)	(in)	(in)	(in^2/in)	(in)	(psi)	(lb/in)	(in-lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/ in)	(lb/in)	(lb/in)
	34.5	4.703	32.75	0.1092	4.703											
Subcase 1: No bolt preload						100	1638	13407	0	1638	0	0	0	0	0	1
						550	9006	73738	0	9006	0	0	0	0	2	2
						1000	16375	134070	0	16375	0	0	0	0	3	9
Subcase 2: Bolt preload set by normal load						100	1638	13407	1638	1638	1122	1166	1133	1131	897	886
						550	9006	73738	1638	9006	0	182	131	118	0	8
						1000	16375	134070	1638	16375	0	0	0	0	0	0
Subcase 3: Preload betwn normal and accid. load						100	1638	13407	9006	9006	43	28	0	0	73	0
						550	9006	73738	9006	9006	6170	6412	6231	6220	4884	4840
						1000	16375	134070	9006	16375	4929	5428	5230	5208	3895	3682
Subcase 4: Bolt preload set by accident load						100	1638	13407	16375	16375	0	0	0	0	21	41
						550	9006	73738	16375	16375	5091	5274	4963	4953	2836	2800
						1000	16375	134070	16375	16375	11219	11659	11329	11310	8967	8875

Table III.3 (continued)

Case 2: Higher-strength lid material (347 stainless steel, minimum yield strength = 30000 psi, tensile strength = 75000 psi)
Lower-strength bolt material (SA 193, minimum yield strength = 75000 psi, tensile strength = 100000 psi)

Closure lid thickness determined by accident load and allowable bending stress (60000 psi)
Bolt area determined by accident load and allowable tensile stress (75000 psi), no prying is considered
Accident load, 1000 psi; Normal load, 100 psi

Closure lid		Closure bolt								Predicted additional prying tensile bolt force per unit length of b.c.					
										Present simpl. methods		Finite element method			
										Bolt bending not included			Bolt bending included		
Edge radius	Thickness	Bolt circle radius	Area/unit length of b.c.	Length	Pressure load	Fixed-edge force, Ff per unit length of b.c.	Fixed-edge moment, Mf per unit length of b.c.	Bolt preload per unit length of b.c.	Non-prying bolt force per unit length of b.c.	Plate-ring model	Plate-plate model	GEMINI model 1	GEMINI model 2	NIKE model 1	NIKE model 2
										Rigid wall	Rigid wall	Rigid wall	Rigid wall	Flex. wall	Flex. wall
(in)	(in)	(in)	(in^2/in)	(in)	(psi)	(lb/in)	(in-lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/ in)	(lb/in)	(lb/in)
34.5	4.29	32.75	0.21833	4.29											
Subcase 1: No bolt preload					100	1638	13407	0	1638	1443	1535	1419	1408	928	867
					550	9006	73738	0	9006	7937	8441	7806	7742	5108	4802
					1000	16375	134070	0	16375	14430	15347	14194	14075	9290	8702
Subcase 2: Bolt preload set by normal load					100	1638	13407	1638	1638	2538	2607	2454	2445	1763	1735
					550	9006	73738	1638	9006	9032	9513	8842	8778	5938	5602
					1000	16375	134070	1638	16375	15525	16419	15228	15113	10147	9502
Subcase 3: Preload betwn normal and accid. load					100	1638	13407	9006	9006	97	62	0	0	208	114
					550	9006	73738	9006	9006	13959	14337	13499	13445	9581	9225
					1000	16375	134070	9006	16375	20453	21243	19885	19780	13699	13095
Subcase 4: Bolt preload set by accident load					100	1638	13407	16375	16375	0	0	0	0	155	0
					550	9006	73738	16375	16375	11518	11792	10788	10743	6065	5682
					1000	16375	134070	16375	16375	25381	26067	24544	24446	17304	16700

Table III.3 (concluded)

Case 3: Higher-strength lid material (347 stainless steel, minimum yield strength = 30000 psi, tensile strength = 75000 psi)
Lower-strength bolt material (SA 193, minimum yield strength = 75000 psi, tensile strength = 100000 psi)

Closure lid thickness determined by accident load and allowable bending stress (60000 psi)
Bolt area determined by 2.5 times accident load and allowable tensile stress (75000 psi), no prying is considered
Accident load, 1000 psi; Normal load, 100 psi

Closure lid		Closure bolt								Predicted additional prying tensile bolt force per unit length of b.c.					
										Present simpl. methods		Finite element method			
										Bolt bending not included			Bolt bending included		
Edge radius	Thick-ness	Bolt circle radius	Area/unit length of b.c.	Length	Pressure load	Fixed-edge force, Ff per unit length of b.c.	Fixed-edge moment, Mf per unit length of b.c.	Bolt preload per unit length of b.c.	Non-prying bolt force per unit length of b.c.	Plate-ring model	Plate-plate model	GEMINI model 1	GEMINI model 2	NIKE model 1	NIKE model 2
										Rigid wall	Rigid wall	Rigid wall	Rigid wall	Flex. wall	Flex. wall
(in)	(in)	(in)	(in^2/in)	(in)	(psi)	(lb/in)	(in-lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/in)	(lb/ in)	(lb/in)	(lb/in)
34.5	4.29	32.75	0.545825	4.29											
Subcase 1: No bolt preload					100	1638	13407	0	1638	3507	3587	3285	3252	1834	1705
					550	9006	73738	0	9006	19290	19728	18068	17886	10193	9693
					1000	16375	134070	0	16375	35073	35868	32850	32521	18516	17610
Subcase 2: Bolt preload set by normal load					100	1638	13407	1638	1638	4239	4293	3952	3924	2409	2468
					550	9006	73738	1638	9006	20021	20434	18735	18558	10708	10185
					1000	16375	134070	1638	16375	35804	36575	33517	33192	19105	18084
Subcase 3: Preload between normal and accid. lo					100	1638	13407	9006	9006	162	102	0	0	3760	311
					550	9006	73738	9006	9006	23313	23612	21735	21581	14917	12598
					1000	16375	134070	9006	16375	39096	39753	36517	36214	22533	20482
Subcase 4: Bolt preload set by accident load					100	1638	13407	16375	16375	0	0	0	0	6439	99
					550	9006	73738	16375	16375	19236	19421	17366	17234	12760	8958
					1000	16375	134070	16375	16375	42388	42930	39517	39237	27136	23195

Table III.4 Finite Element Models Used for the Study of Prying Action and the Verification of the Simplified Analysis Methods

Model identification	Purpose of model	Assumptions and finite elements used
GEMINI 1	To verify the simplified models developed in Appendix III for prying analysis.	Same assumptions are used as in the simplified models for prying analysis (i.e., the closure lid behaves as an elastic plate and is modeled using 3D plate elements, the bolt resists only tension and is modeled with a linear spring element, the cask wall is rigid and is represented only as rigid supports or constraints.)
GEMINI 2	To study the interaction of bolt bending and prying.	Same as Model GEMINI 1, except a rotational spring is added to model the bending stiffness of the closure bolt.
GEMINI 3	To verify the simplified model developed in in Appendix IV for bending analysis.	Same assumptions are used as in the simplified model for bending analysis. The model is identical to the model GEMINI 1, except the bolt is a rotational spring.
NIKE 1	To study the prying and bending effects using a more realistic and detailed finite element model of the bolted joint including the cask wall elasticity.	Both the closure lid and bolts are modeled using 2D axisymmetric solid elements. The bolts are modeled as an equivalent circular cylinder with same tensile and bending stiffness as the assembly of bolts. The bolt model has same area and length as the bolts but an artificial elastic modulus. The bolt elements are connected to the lid at the top and to the cask wall at the bottom. The contact area between the lid and the cask wall is modeled as a sliding interface with friction.
NIKE 2	To study the transmission of force and moment between the closure lid and the bolt head.	Same model as NIKE 1, except the bolt head is explicitly modeled here. The interface between the bolt head and the lid is modeled as a sliding interface with friction.

APPENDIX IV

Maximum Bolt Bending Moment

1.0 Introduction

This appendix continues the analysis of the bolt prying and bending actions of a load applied on the closure lid. The analysis begins in Appendix III with a study of the prying action, and this appendix completes the analysis with the investigation of the bending action. Specifically, this appendix develops a simple model to provide an approximate estimate of the maximum bolt bending moment that an applied load can generate. The model ignores the prying action completely and assumes that only the closure bolts resist the bending of the closure lid. Without the contribution from the prying action, the result obtained from this model for the bolt bending moment is definitely conservative. The analysis of the model produces a simple, closed-form formula for the calculation of the maximum bending bolt moment. The formula, which is used in Table 2.2, shows that the bolt moment is determined by the bolt and closure-lid stiffnesses and by the applied load. This appendix also presents test data and finite element results demonstrating the adequacy of the simplified analysis model and formula. Although the present method produces results that compared reasonably well with test data for an automobile piston cap, the method appears to generate over-conservative results for the shipping cask. This conservatism is probably due to the assumption of a rigid connection between the bolt and the closure lid. In reality, the lid material deforms under the bolt head, and the bolt head does not rotate with the closure lid like a rigid joint. The finite element analysis using the NIKE computer program shows that the bending stress in the bolt is usually less than 20% of the average tensile stress in the closure bolt. Therefore, the bending stress is not likely to cause bolt failures with gross plastic deformation. However, its influence on bolt failures with incremental plastic deformation and fatigue can still be significant. A thicker closure lid will reduce the bending bolt moment and stress. A higher preload is not likely to reduce the bending moment appreciably, but it will reduce the significance of the bending stress in fatigue.

2.0 Analysis

To obtain the most conservative estimate of the bending bolt moment, the following assumptions are made in the analysis: (1) the cask wall is rigid; (2) there exists no prying action, and the rotation of the closure lid relative to the cask wall is resisted only by the bending of the closure bolts. Furthermore, the closure lid is treated as a plate, the bolt as a beam, and the lid and bolt are rigidly connected at the bottom of the bolt head.

Figure IV.1 shows the analytical model and submodels used to find the bending bolt moment, which is identified as Mb in the figure. Similar to the approach used for the analysis of prying in Appendix III, the applied force is represented by a fixed-edge force (Ff) and a fixed-edge moment (Mf) applied at the bolt circle. Using formulas from Reference IV.1 for the beam and plate, the continuity of moment and rotation across the beam-plate boundary can be expressed as follows:

$$Ml = Mb \qquad \text{(IV.1)}$$

$$\theta l = \theta b \qquad \text{(IV.2)}$$

where Ml is the plate (lid) radial bending moment at the bolt circle; θp is the plate (lid) rotation at the bolt circle and about the beam (bolt) circle; and θb is the bolt rotation at the bolt head about the bolt circle as follows:

$$\theta l = \frac{1}{kl}(Mf - Ml) \qquad \text{(IV.3)}$$

$$\theta b = \frac{1}{kb} Mb \qquad \text{(IV.4)}$$

where

$$kl = \frac{E\,t^3}{6\left[(1-v^2) + (1-v)^2\left(\frac{b}{c}\right)^2\right] b}\ ; \qquad \text{(IV.5)}$$

$$kb = \frac{Eb\ Ib}{Lb}\ ; \qquad \text{(IV.6)}$$

E, v, and t are the Young's modules, Poissons' ratio and thickness of the closure lid, respectively; Eb, Lb, and Ib are the Young's modules, length, bending moment of inertia of the closure bolt, respectively; b is the radius of the bolt circle. Similar to the moments, the bolt moment of inertia is defined for a unit length of the bolt circle as follows:

$$Ib = \frac{Nb\ I}{2\,\pi\,R} \qquad \text{(IV.7)}$$

where Nb is the total number of bolts; I is the moment of inertia of the bolt cross-section about a bolt diameter:

$$I = \frac{\pi\, Db^4}{64} \qquad \text{(IV.8)}$$

where Db is the nominal diameter of the bolt.

Equations IV.1 and IV.2 can be solved for the bolt bending moment:

$$Mb = \frac{kb}{kb + kl} Mf \qquad \text{(IV.9)}$$

This equation indicates that the bolt bending moment is only a fraction of the applied moment (Mf) and this fraction decreases with decreasing bolt stiffness relative to the closure lid. Therefore, a thicker closure lid will reduce bolt bending as well as bolt prying. The effect of plate thickness on the prying action is discussed in Appendix III.

3.0 Verification of Analysis

The validity of the preceding analysis and the resulting simple formula for the evaluation of bending bolt moment is demonstrated in this section by comparing the analysis results first with experimental results and then with finite element analysis results. The agreements of the results are excellent. The results of this section also provide some information concerning the relative magnitudes of bending and average tensile stresses in bolts. For most situations, the average tensile stress governs the bolt design.

The experimental results used for the comparison were obtained by Radzimovsky and Kasuba in 1962 for a bolted connecting rod. The geometry of the test specimen, the loading method, and the strain gage locations are shown in Fig. IV.2. The data used for the comparison was obtained with the maximum clearance between the loading pin and the connecting-rod cap. For this case, the loading exerted by the loading pin on the connecting-rod cap is approximately a concentrated load which is identified as Pe in Fig. IV.2. Considering the cap to be a semi-circular arch or ring, the fixed-edge moment (Mf) and the bending stiffness kl needed for Equation IV.9 can be obtained as follows:

$$Mf = \frac{\pi^2 - 2\pi - 4}{\pi^2 - 8} P R = 0.22 P R \qquad \text{(IV.11)}$$

$$kl = \frac{2 E I}{\pi R} \qquad \text{(IV.12)}$$

where P is one-half of the applied load Pe on the connecting-rod cap; R, E, and I are the mean radius, the Young's modules and the cross-sectional area moment of inertia the connecting-rod cap, respectively. Using these two equations and Equations IV.9, IV.6, and IV.7, the bolt bending moment and corresponding maximum bending stress on the bolt cylindrical surface are obtained and compared in Table IV.1 to the experimental results. Despite the fact that the test results of the average tensile bolt stress show the existence of a prying effect, the test and analysis results of the bending stress compare closely. Thus the validity of the present analysis approach is demonstrated.

To further demonstrate the usefulness of the present analysis and to obtain information concerning the significance of bending bolt stress in reality, Equations VI.9, IV.5, IV.6, and III.45 for the evaluation of bending bolt moment in a bolted closure are applied to the three bolted closure designs described in Appendix III for a typical rail cask. The bending moment results are presented in Tables IV.2 with similar results from finite element analyses. The finite element models used for the analyses are described in Appendix III. The GEMINI finite element model 3 is equivalent to the present simplified model for bending evaluation (i.e., the rotation of the closure lid in the model is resisted only by the bending action of the bolts but not by the prying action). Accordingly, the results of these two analyses should compare closely. The results in Tables IV.2 indeed confirm this expectation. The GEMINI model 2 also includes the prying action, which should help reduce the bolt bending. However, the comparison of the results of the two GEMINI models show only small reductions in the bolt bending due to the bolt prying. The NIKE models predict much lower bending bolt moment than the GEMINI models and the simplified model. This is probably due to the deformation of the lid material under the bolt head, which is only modeled in the NIKE models. The reduction of the bending bolt moment cannot be caused by the modeling of the interface between the bolt head and the lid because this interface is modelled differently in the two NIKE models and the models show insignificant difference in the bending bolt moment. The NIKE results also show that the bending stress is less than 20% of the average tensile bolt force.

In comparison, the prying force can be more than 60% of the average tensile bolt force. Therefore, the prying is a much more significant effect than bending.

4.0 Maximum Bending Bolt Moment Generated by Inward Load

In the foregoing discussion of Subsection 2.0 of this appendix, the applied force is always directed towards the cask exterior. However, the same results can also be used, to determine the maximum bending moment generated by an inward load (i.e., a load directed toward the cask interior). The difference between the maximum bending actions generated by the inward and the outward load will be insignificant if the difference between the diameters of the bolt circle and the cask cavity is small compared to the diameter of the bolt circle. In the case of the outward load, the bending action is caused by the applied load and the bolt force at the bolt circle, while in the inward load case, the action is generated by the applied load and the cask-wall reaction at the cask cavity.

References (Appendix IV)

IV.1 W. C. Young, *Roark's Formulas for Stress and Strain*, 6th Ed., McGraw Hill, New York, NY, 1989.

IV.2 E. Radzimovsky and R. Kasuba, "Bending Stresses in the Bolts of a Bolted Assembly," *Experimental Mechanics*, Vol. 2, No. 9, September 1962.

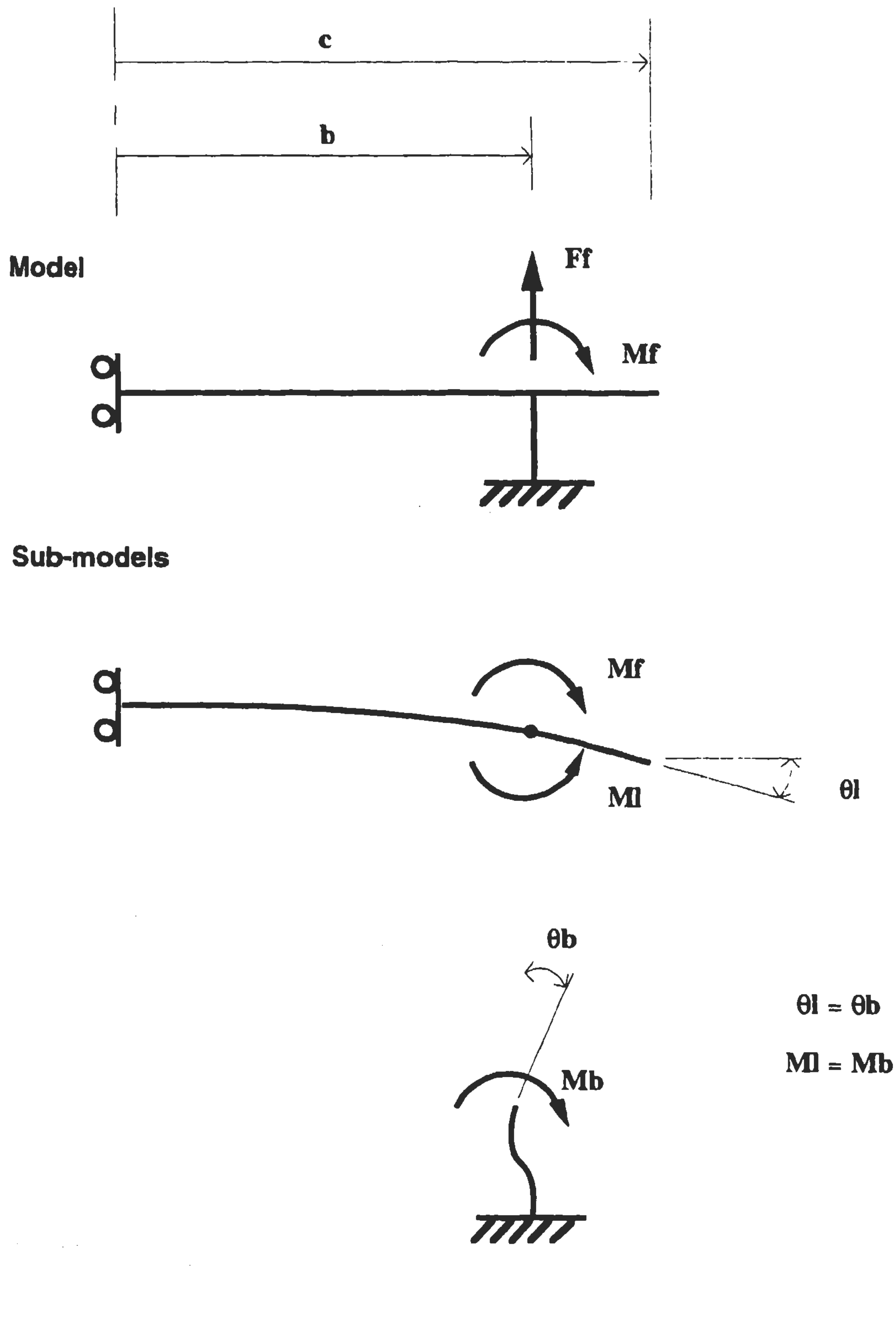


Figure IV.1 Analytical model and sub-models for evaluating bending bolt moment.

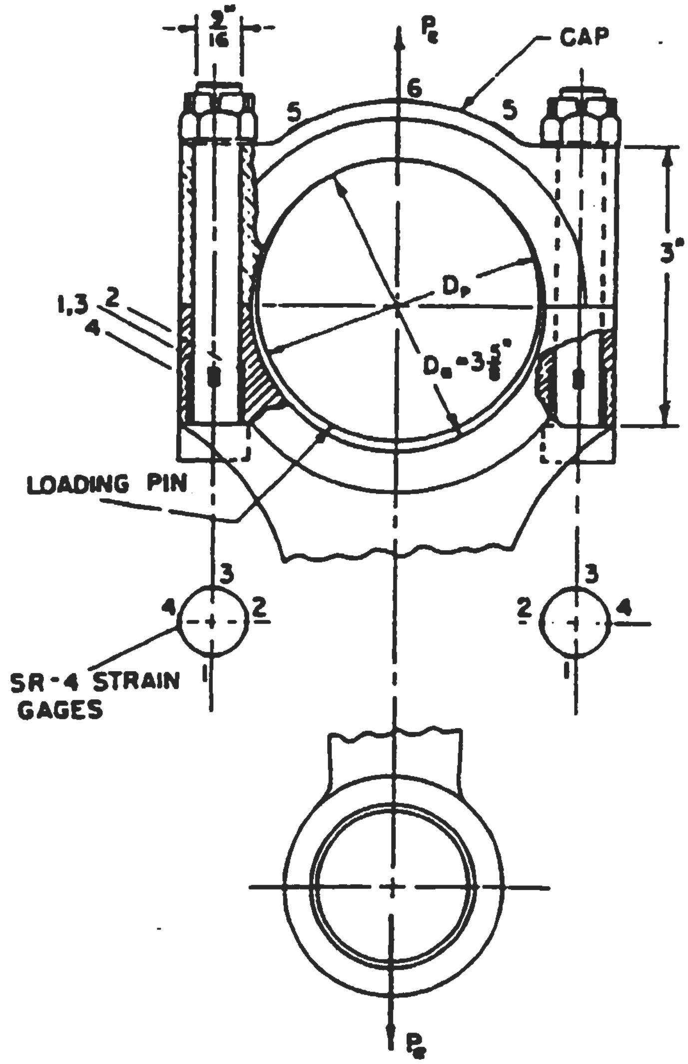


Figure IV.2 Bolted connecting rod cap for comparing the results of test and analysis.

Table IV.1 Comparison of Analytical and Experimental Results of Bolt Bending Stress in Connecting-rod Cap

Notes:

Analytical results were obtained using the method described in Appendix IV of this report
Experimental results were measured by Radzimovsky & Kasuba (Reference IV.2)

Bolt stresses include contribution from bolt preload.

Test results for average axial and maximum bending stresses are from strain gages 3 and 2, respectively.

Difference between the analytical and experimental results for the average axial stress is attributable to the prying effect.

Connecting-rod cap dimensions				Bolt dimensions				Analytical results							Experimental results		
								Model information		Bolt force/moment		Bolt stresses			Bolt stresses		
Width (in)	Thickness (in)	Radius (in)	Cap-pin clearance (DB-DP)/2 (in)	Diameter (in)	Length (in)	Bolt preload (lb)	Total applied load, Pe (lb)	Fixed-edge moment, Mf (in-lb)	Bolt length, Lb (in)	Axial force (lb)	Bending moment (in-lb)	Pre-stress (psi)	Average axial (psi)	Maximum bending (psi)	Pre-stress (psi)	Average axial (psi)	Maximum bending (psi)
1.00	0.75	1.95	0.011	0.56	3.00	6000	20000	4297	1.5	10000	955	24144	40241	54676	24000	51500	55500

Table IV.2 Bending Bolt Moment and Stress in Sample Rail Cask Closure Designs

	Closure lid: Edge radius (in)	Closure lid: Thick-ness (in)	Closure bolt: Bolt circle radius (in)	Closure bolt: Area/unit length of b.c. (in^2/in)	Closure bolt: Length (in)	Press. load (psi)	Bolt preload per unit length of b.c. (lb/in)	Non-prying bolt force per unit length of b.c. (lb/in)	Bolt bending moment per unit length of b.c. (in-lb/in): No prying: Present simpl. method	With prying: Finite element method: GEMINI model 3	With prying: Finite element method: GEMINI model 2	With prying: Finite element method: NIKE model 1	With prying: Finite element method: NIKE model 2	Predicted bolt stress (including prying) for closure bolts of 3/4" dia.: Present simplified method: Plate-ring model: Average axial stress (psi)	Present simplified method: Plate-plate model: Average axial stress (psi)	Present simplified method: Plate-plate model: Maximum bending stress (psi)	Finite element: NIKE model 2: Average axial stress (psi)	Finite element: NIKE model 2: Maximum bending stress (psi)
Case 1: Lower-strength lid material (304 stainless steel, minimum yield strength = 25000 psi, tensile strength = 65000 psi); Higher-strength bolt material (SA 540, minimum yield strength = 150000 psi, tensile strength = 165000 psi). Closure lid thickness determined by accident load and allowable bending stress (60000 psi). Bolt area determined by accident load and allowable tensile stress (150000 psi), no prying is considered. Accident load, 1000 psi; normal load, 100 psi																		
	34.5	4.703	32.75	0.1092	4.703													
Subcase 1: No bolt preload						100	0	1638	28	28	28	16	15	14995	14995	2724	15003	1434
						550	0	9006	153	152	152	79	81	82475	82475	14980	82492	7960
						1000	0	16375	279	277	277	142	142	149954	149954	27237	150034	13874
Subcase 2: Bolt preload set by normal load						100	1638	1638	28	28	28	11	13	25270	25673	2724	23106	1235
						550	1638	9006	153	152	152	79	79	82475	84141	14980	82546	7689
						1000	1638	16375	279	277	277	142	144	149954	149954	27237	149980	14079
Subcase 3: Preload betwn normal and accid. loads						100	9006	9006	28	28	28	16	10	82869	82731	2724	83142	940
						550	9006	9006	153	152	152	84	63	138977	141193	14980	126825	6117
						1000	9006	16375	279	277	277	149	143	195092	199661	27237	182830	13976
Subcase 4: Bolt preload set by accident load						100	16375	16375	28	28	28	4	6	149954	149954	2724	150251	547
						550	16375	16375	153	152	152	96	82	196575	198251	14980	173758	8013
						1000	16375	16375	279	277	277	130	146	252692	256722	27237	229628	14267

Table IV.2 (continued)

Case 2:

Higher-strength lid material (347 stainless steel, minimum yield strength = 30000 psi, tensile strength = 75000 psi)
Lower-strength bolt material (SA 193, minimum yield strength = 75000 psi, tensile strength = 100000 psi)

Closure lid thickness determined by accident load and allowable bending stress (60000 psi)
Bolt area determined by accident load and allowable tensile stress (75000 psi), no prying is considered
Accident load, 1000 psi; normal load, 100 psi

	Closure lid		Closure bolt						Bolt bending moment per unit length of b.c. (in-lb/in)					Predicted bolt stress (including prying) for closure bolts of 3/4" dia.				
									No prying	With prying				Present simplified method			Finite element	
										Finite element method				Plate-ring model	Plate-plate model		NIKE model 2	
	Edge radius (in)	Thickness (in)	Bolt circle radius (in)	Area/unit length of b.c. (in²/in)	Length (in)	Press. load (psi)	Bolt preload per unit length of b.c. (lb/in)	Non-prying bolt force per unit length of b.c. (lb/in)	Present simpl. method	GEMINI model 3	GEMINI model 2	NIKE model 1	NIKE model 2	Average axial stress (psi)	Average axial stress (psi)	Maximum bending stress (psi)	Average axial stress (psi)	Maximum bending stress (psi)
	34.5	4.29	32.75	0.21833	4.29													
Subcase 1: No bolt preload						100	0	1638	80	79	64	33	34	14201	14531	3919	11469	1662
						550	0	9006	441	432	351	208	183	77604	79912	21552	63244	8920
						1000	0	16375	802	786	637	385	332	141094	145417	39185	114856	16214
Subcase 2: Bolt preload set by normal load						100	1638	1638	80	79	64	30	27	19125	19441	3919	15449	1326
						550	1638	9006	441	432	351	232	214	82619	84822	21552	66911	10434
						1000	1638	16375	802	786	638	458	373	146109	150204	39185	118523	18237
Subcase 3: Preload betwn normal and accid. loads						100	9006	9006	80	79	64	36	33	41695	41535	3919	41772	1625
						550	9006	9006	441	432	351	242	274	105186	106917	21552	83504	13379
						1000	9006	16375	802	786	638	366	359	168680	172299	39185	134981	17551
Subcase 4: Bolt preload set by accident load						100	16375	16375	80	79	64	20	15	75001	75001	3919	75001	745
						550	16375	16375	441	432	351	191	230	127756	129011	21552	101027	11255
						1000	16375	16375	802	786	638	358	388	191252	194394	39185	151493	18948

Table IV.2 (concluded)

	Closure lid: Edge radius (in)	Closure lid: Thickness (in)	Closure bolt: Bolt circle radius (in)	Closure bolt: Area/unit length of b.c. (in²/in)	Closure bolt: Length (in)	Press. load (psi)	Bolt preload per unit length of b.c. (lb/in)	Non-prying bolt force per unit length of b.c. (lb/in)	Bolt bending moment per unit length of b.c. (in-lb/in): No prying: Present simpl. method	With prying: Finite element method: GEMINI model 3	With prying: Finite element method: GEMINI model 2	With prying: Finite element method: NIKE model 1	With prying: Finite element method: NIKE model 2	Predicted bolt stress (including prying) for closure bolts of 3/4" dia.: Present simplified method: Plate-ring model: Average axial stress (psi)	Present simplified method: Plate-plate model: Average axial stress (psi)	Present simplified method: Maximum bending stress (psi)	Finite element: NIKE model 2: Average axial stress (psi)	Finite element: NIKE model 2: Maximum bending stress (psi)
Case 3: Higher-strength lid material (347 stainless steel, minimum yield strength = 30000 psi, tensile strength = 75000 psi) Lower-strength bolt material (SA 193, minimum yield strength = 75000 psi, tensile strength = 100000 psi)																		
Closure lid thickness determined by accident load and allowable bending stress (60000 psi) Bolt area determined by 2.5 times accident load and allowable tensile stress (75000 psi), no prying is considered Accident load, 1000 psi; normal load, 100 psi																		
	34.5	4.29	32.75	0.545825	4.29													
Subcase 1: No bolt preload						100	0	1638	199	195	111	63	84	9425	9572	3884	6124	1645
						550	0	9006	1093	1072	612	412	501	51841	52644	21360	34259	9793
						1000	0	16375	1987	1948	1113	847	859	94257	95714	38837	62263	16790
Subcase 2: Bolt preload set by normal load						100	1638	1638	199	195	111	55	52	10766	10865	3884	7522	1013
						550	1638	9006	1093	1072	612	394	370	53181	53937	21360	35159	237
						1000	1638	16375	1987	1948	1113	990	872	95597	97009	38837	63131	17037
Subcase 3: Preload between normal and accid. load						100	9006	9006	199	195	111	33	59	16797	16687	3884	17070	1158
						550	9006	9006	1093	1072	612	353	330	59212	59760	21360	39580	6448
						1000	9006	16375	1987	1948	1113	832	884	101628	102831	38837	67526	17270
Subcase 4: Bolt preload set by accident load						100	16375	16375	199	195	111	12	23	30000	30000	3884	30183	440
						550	16375	16375	1093	1072	612	344	380	65243	65581	21360	46412	7434
						1000	16375	16375	1987	1948	1113	777	709	107659	108652	38837	72524	13848

APPENDIX V

Maximum Non-prying Tensile Bolt Force Caused By Impact Load

This appendix develops the formulas given in Tables 4.5 and 4.6 for the calculation of non-prying tensile bolt force generated by an impact load. The development is based on the assumption that during an oblique impact, the closure lid experiences a rigid body rotation about the impact point. This rotation, in turn, causes the closure bolts to be stretched in different amounts proportional to the distance between the bolts and the axis of rotation as depicted in Fig. V.1.

The maximum bolt elongation and tensile bolt force occur in the bolt which is located at the farthest distance from the impact point. This maximum bolt force can be obtained by considering the closure lid as a lever with its fulcrum located at the impact point. The condition for the determination of the bolt forces is that the total moment of the bolt forces about the impact point must be equal to the total moment of the impact load about the same point:

$$\text{Sum of } (fb\ yb) = L\ yL \qquad (V.1)$$

where fb and yb are the axial bolt force and its distance from the impact point, respectively; similarly, L and yL are the impact load and its distance from the impact point, respectively.

The bolt force fb is equal to the product of the axial stiffness and elongation of the bolt:

$$fb = \frac{Ab\ Eb}{Lb}\ ub \qquad (V.2)$$

where Ab, Eb, ub, and Lb are the cross-sectional area, Young's modulus, axial elongation, and length of the bolt. The axial elongation is directly proportional to the distance of the bolt from the impact point:

$$ub = \frac{yb}{yr}\ ur \qquad (V.3)$$

where ur and yr are the displacement and distance of a reference point, respectively. Inserting Equations V.2 and V.3 into Equation V.1, the following equation is obtained for the solution of the bolt force (fb):

$$\frac{Eb\ ur}{Lb\ yr}\ \text{sum of } (Ab\ yb^2) = L\ yL \qquad (V.4)$$

where the sum of ($Ab\ yb^2$) is the area moment of inertia (I) of the assembly of bolts about the impact point. Using the parallel-axis theorem, this quantity can be expressed in terms of the moment of inertia about the centroid of the bolt assembly ($\bar{I}$):

$$\text{sum of } (Ab\ yb^2) = I = \bar{I} + A\ \bar{y}^2 \qquad (V.5)$$

where A is the total area of the bolt assembly; y is the distance between the impact point and the centroid of the bolt assembly (i.e., the center of the closure lid).

Combining Equations V.2 through V.5, the following general formula is obtained for the evaluation of the bolt force fb:

$$fb = \frac{L\,yL}{\bar{I} + A\,\bar{y}^{2}}\,Ab\,yb \qquad (V.6)$$

It can be shown that the moment of inertia of the bolt assembly can be obtained by considering the bolt assembly to be a very thin circular annulus having the same radius as the bolt circle and the same area as the bolt assembly. Using the formula from Reference V.1 for the thin ring, an expression can be obtained for $\bar{I}$:

$$\bar{I} = \frac{A\,Rlb^{2}}{2} \qquad (V.7)$$

where Rlb is the bolt-circle radius, A is the total bolt area equal to the single bolt area (Ab) multiplied by the total number of bolts (Nb):

$$A = Nb\,Ab \qquad (V.8)$$

Ignoring the small difference between the radii of the bolt circle and the closure lid (i.e., Rlb = Rlo) and assuming the impact point to be at the edge of the closure lid (i.e., yL = Rlo and $\bar{y}$ = Rlo) Equations V.6, V.7, and V.8 can be combined to give the following formula for the bolt force of a bolt at yb:

$$fb = \frac{2}{3}\left(\frac{L}{Nb}\right)\frac{yb}{Rlo} \qquad (V.9)$$

From this formula, the maximum bolt force, or the bolt force of the bolt located at the farthest distance from the impact point (yb = 2 Rlo) can be obtained:

$$(fb)_{max} = \frac{4}{3}\left(\frac{L}{Nb}\right) \qquad (V.10)$$

The quantity L/n represents the average axial bolt force which would be the bolt force magnitude if the closure lid were not impacted at the edge. Thus, impacting at the edge of the closure lid can raise the bolt force by as much as 34%. For conservatism, this higher magnitude is used for all the oblique impact conditions depicted in Tables 4.5 and 4.6.

Reference (Appendix V)

V.1 W. C. Young, *Roark's Formulas for Stress and Strain*, 6th Ed., McGraw Hill, New York, NY, 1989.

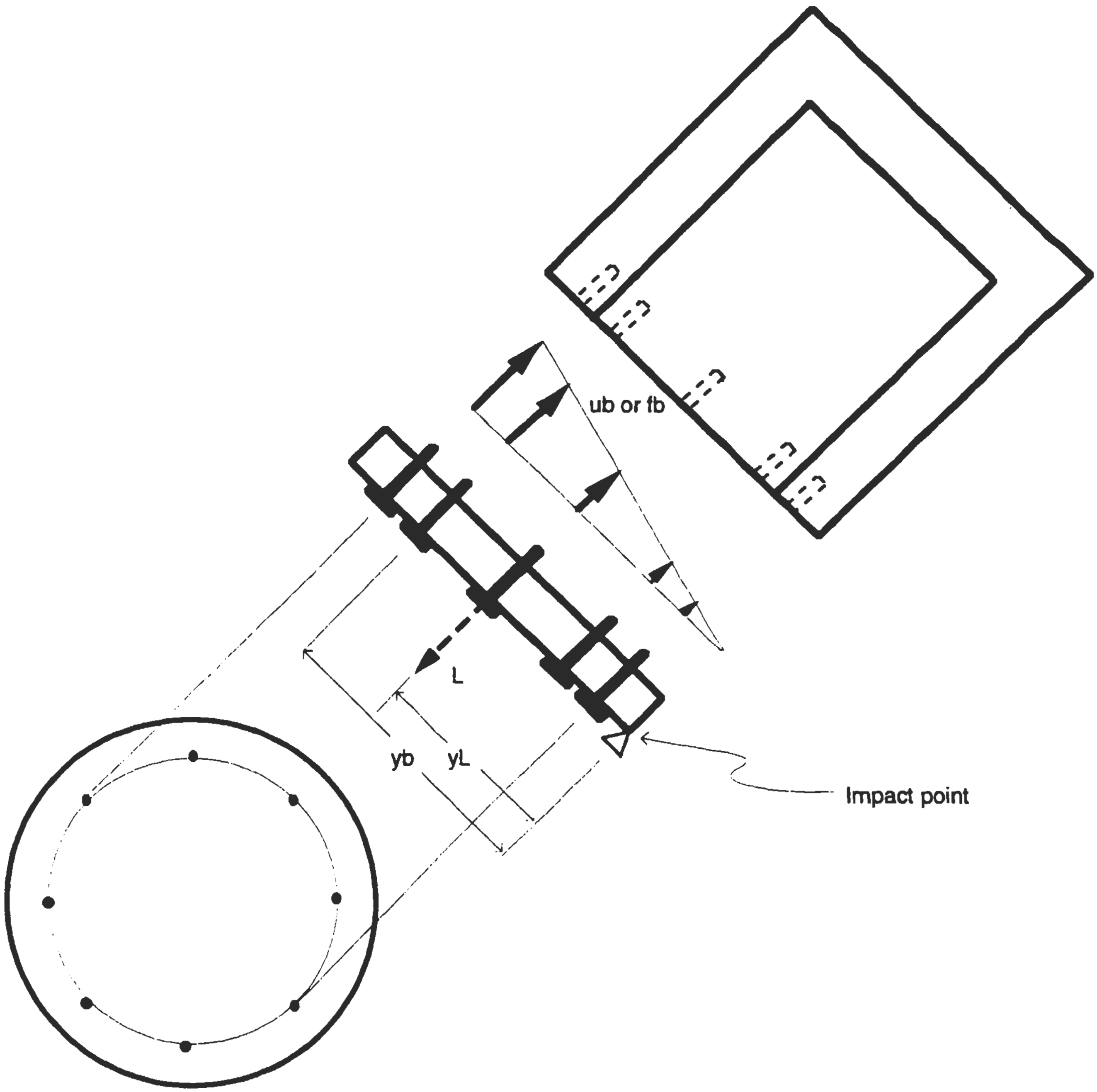


Figure V.1 Axial elongation and force of closure bolts generated by a rigid closure lid during oblique impact.

APPENDIX VI

Maximum Puncture Load

This appendix develops the approximate formula given in Table 4.7 of this report for the calculation of the maximum puncture force (PUN). The formula is the result of an approximate analysis of the puncture process of a plate in the direction normal to the closure lid surface. The force required to puncture the lid is compared to the force required to indent the lid, and the smaller of these two forces is used as the approximate estimate of the maximum puncture force.

In the analysis, the puncture bar is considered to be rigid. This is a justifiable assumption because the puncture bar is usually much stronger than the closure lid. Therefore, the puncture force is determined only by the closure-lid deformation. There are two possibilities—either the closure lid is indented without puncture or it is also punctured. If it is punctured, the puncture force will be determined by the puncture resistance of the closure lid. On the other hand, if the lid is only plastically deformed, the maximum force will be determined by the indentation resistance of the closure lid. The lower of these two estimates of the puncture force can be used as the puncture load (PUN) for the closure bolt analysis:

$$PUN = \text{Min.}\,(\, PUNP,\ PUNI \,) \qquad (VI.1)$$

where PUNP and PUNI are the puncture forces determined from the analyses of the puncture and indentation processes, respectively. Approximate formulas are developed in the remainder of this appendix for the estimate of these puncture forces.

In the case of puncture, the failure can be approximately modeled as a shear-plug failure along the edge of the puncture bar cross-section. Figure VI.1 shows this simplified failure mode. Mok (Ref. VI.1) has shown that this model can provide a correlation among the puncture energy, puncture-bar diameter, and plate thickness which is similar to an empirical relation obtained by Lo (Ref. VI.2) from existing puncture test results of steel plates. The puncture force (PUN) for this failure mode can be calculated as the product of the shear-plug edge area (As) and the ultimate shear stress (Sf) of the lid material:

$$PUNP = As\ \ Sf \qquad (VI.2)$$

where

$$As = \pi\, Dpb\ tl \qquad (VI.3)$$

$$Sf = 0.6\, Su \qquad (VI.4)$$

Dpb is the diameter of the puncture bar; tl is the closure lid thickness; and Su is the ultimate tensile stress of the lid material at the puncture temperature. The relation used here, Equation VI.4 between the ultimate shear stress (Sf) and the ultimate tensile stress (Su), is a widely accepted empirical relation.

In the case of no puncture, a conservative estimate of the maximum impact force can be obtained with the assumption that the puncture force is sufficient to produce on the closure lid, a permanent indentation whose diameter is equal to the puncture-bar diameter. Thus, the puncture force can be estimated as the product of the indentation pressure and area. Past analytical and experimental studies of the indentation of a soft metal by a rigid punch have shown that the indentation pressure (Pi) is directly proportional to the uniaxial yield stress (Syl) of the indented material (Refs. VI.3 through VI.5):

$$\pi = c\ Syl \tag{VI.5}$$

where the coefficient c of Syl is a constant having an approximate value of 3. For a work-hardening material, the yield stress Syl used in this relation is dependent on the indentation depth, which is a measure of the average permanent strain over the indentation area. However, for the present analysis, it is assumed that most of the impact energy is spent on bending and puncturing the closure lid and only a shallow indentation is produced. Therefore, the work-hardening effect is ignored and the 0.2 % off-set yield stress (Sy) of the closure-lid material is used as Syl. Accordingly, the maximum indentation force (PUNI) is calculated as follows:

$$PUNI = .25\ c\ \pi\ Dpb^2\ Sy \tag{VI.6}$$

Equations VI.1, VI.3, and VI.6 provide all of the formulas needed for a conservative estimate of the puncture load at all impact angles.

The puncture process at an oblique impact angle is quite different from the normal impact. Ho[illegible] maximum impact force is concerned, it is conservative to use the normal impact results [illegible] this basis, the maximum puncture forces of an oblique impact in the directions normal and tangential to the closure lid can be obtained as follows:

$$Ln = PUN \sin\theta \tag{VI.7}$$

(VI.8)

where PUN is the maximum puncture force of the normal impact from Equation VI.1; θ is the angle between the closure lid surface and the impact direction; and Ln and Lt are the puncture loads normal and tangential to the closure lid, respectively.

References

VI.1 G. C. Mok, "SCANS Workshop Course Material," Lawrence Livermore National Laboratory, University of California, May 1991.

VI.2 T.Y. Lo, "Puncture of Shipping Casks-SCANS Theory Manual, Vol. 7," U.S. Nuclear Regulatory report NUREG/CR-4554, Vol. 7, February 1990.

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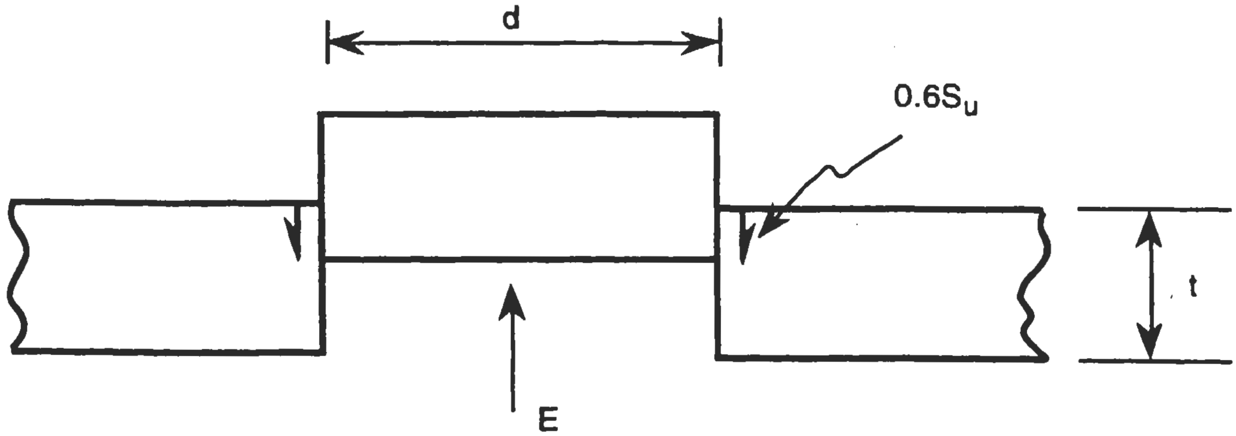

Assumption:

- Puncture by shear rupture at edge of punch area of diameter d
- Shear rupture occurring at constant stress determined by the ultimate tensile strength, S_u
- Puncture energy, E = Work to produce shear plug

Result: $$\frac{t}{d} \propto \left(\frac{E}{S_u\, d^3} \right)^{0.5}$$

Figure VI.1 Shear-plug failure mode for evaluation of puncture force.

APPENDIX VII

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Section III, Rules for Construction of Nuclear Power Plant Components Division 1
Subsection NB, Class 1 Components; Subsection NC, Class 2 Components
Mandatory Appendices:

Appendix I,	Design Stress Intensity Valued, Allowable Stresses Material Properties, and Design Fatigue Curves
Table I-1.3	Design Stress Intensity Values Sm for Bolting Materials (Class 1 Components)
Table I-7.3	Allowable Stress Values S for Bolting Materials for Class 2 and 3 Components
Table I-8.3	Allowable Stress Values S for Bolting Materials for Class 3 Components
Figure I-9.0	Design Fatigue Curves (for bolts having tensile strength < 100ksi)
Figure I-9.4	Design Fatigue Curve for High-Strength Steel Bolting for Temperature Not Exceeding 700F (for bolts having tensile strength > 100ksi)
Appendix III,	Basis for Establishing Design Stress Intensity Values and Allowable Stress Values
Appendix XI,	Rules for Bolted Flange Connections for Class 2 and 3 Components and Class MC Vessels
Appendix XII,	Design Considerations for Bolted Flange Connections
Appendix XIII,	Design Based on Stress Analysis for Vessels. Designed in Accordance with NC-3200. (For Class 2 components, equivalent to design by analysis for Class 1, use Class 1 allowable stress for bolts.)
Appendix XIV,	Design Based on Fatigue Analysis for Vessels. Designed in Accordance with NC-3200 (for Class 2 components).

Non-Mandatory Appendices:

Appendix A,	Stress Analysis Methods (shell discontinuity analysis, interaction equation)
Appendix E,	Minimum Bolt Cross-Sectional Area (design mechanical loads for bolts of Class 1 components)

Appendix F, Rules for Evaluation of Service Loading with Level D Service Limits (for Class 1 components)
Appendix G, Protection Against Nonductile Failure
Appendix L, Class FF Flange Design for Class 2 and 3 Components and Class MC vessels (prying force)
Appendix N, Dynamic Analysis Methods Section VIII, Rule for Construction of Pressure Vessels

Section VIII Rules for Construction of Pressure Vessels
Division 1
Mandatory Appendices:
Appendix 2, Rules for Bolted Flange Connections with Ring-Type Gaskets
Non-Mandatory Appendices:
Appendix S, Design Considerations for Bolted Flange Connections
Appendix Y, Flat-Face Flanges with Metal-to-Metal Contact Outside the Bolt Circle

Section VIII Rules for Construction of Pressure Vessels
Division 2, Alternative Rules
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